HYDRO-ELECTRIC ENGINEERING

Volume I
CIVIL AND MECHANICAL

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PREFACE.

At the present time great interest is being taken in the development of the water-power resources of this and other countries of the world, and it is certain that in the near future a very large amount of hydro-electric development will take place.

Few engineering problems demand a more general knowledge than those presented in the development of a water-power scheme. The engineer has to deal with such diverse matters as rainfall and run-off, gauging operations, the survey and contouring of catchment areas, the construction and foundation of dams, the construction and lay-out of canals, flumes, pressure tunnels, pipe lines and pipe tracks, regulating sluices, and power houses; and with a series of mechanical and electrical problems relating to the arrangement and regulation of hydraulic and electrical machinery, and the installation and operation of transformers, transmission lines, and sub-stations. Since the chance of the commercial development of any scheme depends essentially on the possibility of obtaining a profitable outlet for the power, the engineer should have a sound if not a detailed knowledge of modern electro-chemical, -physical, and -metallurgical processes, and, in general, of the many branches of industry in which cheap electrical power is an essential.

While many of the problems are such as are encountered in other branches of engineering, a number of them are peculiar to this particular branch. Limitations of space make it impossible to attempt to deal, in one treatise, with more than the general principles involved in a successful solution of these problems, and the present treatise has been prepared with this in view. For details of the construction of successful installations, the reader is referred to the excellent articles which appear from time

to time in the technical press, and in the proceedings of the various technical societies. While the illustrative matter in the treatise has been chosen as representing recent practice, each illustration has been chosen as illustrating some definite principle of lay-out, design, or construction.

The work is intended both for the student of the subject and for the designing and operating engineer, and it is hoped that it may prove of some use to such readers. The present volume deals with the preliminary work and with the civil and mechanical side of water-power development. The second volume deals with the electrical equipment and lay-out, with the economical and statistical side of the question, and with the possibilities of tidal power.

In the preparation of Volume I, the editor has had the advantage of the co-operation of Mr. H. D. Cook, M.Sc., B.E., who, in connection with one of the leading British firms of hydro-electric engineers, has had an extensive experience of such developments, and who has contributed Chapters VII and X.

The editor would express his indebtedness to all those who have so kindly assisted in the preparation of this work by suggestions and by the provision of illustrative material. It is hoped that due acknowledgment of such assistance has been made in the body of the text.

A. H. GIBSON.

THE UNIVERSITY,
MANCHESTER, 1921.

CONTRIBUTIONS

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CHAPTER I

Introductory

1. The water-wheel was the first, and for many centuries the most important, agent for utilizing any natural source of energy for industrial work, and largely in consequence of this the early industrial communities grouped themselves around river sites where such power was readily available. The early water-wheels were, however, of crude construction, and were only adapted for very low rotative speeds and small powers, and when the steam-engine was evolved in the eighteenth century, it offered so many advantages over the older form of prime mover as seriously to check water-power development in countries where coal was comparatively cheap. One of the great advantages of the steam-power plant was that it enabled an industry to be developed wherever raw material, fuel, labour, and markets could be most easily assured.

The development of the electric generator during the latter half of the nineteenth century, and the development, since 1880, of high-voltage electric transmission, have, however, indirectly done much towards regaining for water power something of its old relative value in the economic scheme. This tendency has been accelerated in recent years by the ever-increasing cost of coal and oil fuels, and by the growing belief in the urgent necessity for their conservation. The great developments in electro-chemical, -physical, and -metallurgical processes during the past decade have also done much to further the utilization of water powers. Most of these processes require relatively large amounts of energy, and all are economically dependent on the cheapness of this energy. They have created a demand for large blocks of cheap power which can, under favourable circumstances, more readily be satisfied from a water-power installation than from any other source.

Step by step with these developments on the electrical side great advances have been made in the design of the hydraulic prime mover. The first true hydraulic turbine of anything resembling modern form was

evolved about 1823 by M. Fourneyron. Such machines were capable of developing much greater powers, of operating at higher and more convenient speeds, and of more accurate speed regulation than the older water-wheels, which they gradually replaced. In response to the demand of the electrical engineer, the turbine has gradually been improved in design, in efficiency, and in output, resulting in the extremely efficient reaction turbines and Pelton wheels of the present day. Under favourable conditions such units are capable of efficiencies in the neighbourhood of 90 per cent. Turbines are now in operation at Queenstown, Niagara, each of which is capable of developing 60,000 h.p., and it is probable that subsequent units of the same plant will develop 100,000 h.p.

These various developments have made it commercially possible to make use of large water powers at sites far remote from any centre of industrial activity. In many cases industrial communities, attracted by the cheapness of the power, have grown up around such sites. In others the energy has been transmitted electrically for long distances, in some cases between 200 and 300 miles, to some more convenient site. During the last ten years this tendency has been increasingly evident in all those industrial countries naturally favoured with water-power resources, and recent economic developments indicate that even greater attention is likely to be paid to such possibilities in the near future than in the immediate past.

2. Available Water Power.—An estimate by the Dominion Water Power Branch of the Canadian Department of the Interior in 1915, modified by the author in the light of more recent information, outlines the water-power possibilities of the leading countries approximately as follows:

Country.	Area (Square Miles).	Population (Latest available Figures, Pre-War Boundaries).	B. Horse-Power Available.	B. Horse-Power Developed.	Per Cent. Utilized.	Horse-power per Square Mile of Area.	
						Available.	Developed
United States	3,026,600*	92,019,900	28,100,000	7,500,000	26·7	9·3	2·48
Canada ..	2,000,000	8,033,500	22,900,000	3,385,000	14·8	11·4	1·69
Austria-Hungary	241,330	49,418,600	6,460,000	570,000	8·8	26·8	2·37
France ..	207,100	39,601,500	7,500,000	2,000,000	26·7	36·2	9·65
Norway ..	124,130	2,302,700	7,500,000	1,250,000	16·7	60·4	10·10
Spain ..	194,700	18,618,100	5,000,000	540,000	10·8	25·7	2·77
Sweden ..	172,900	5,521,900	6,200,000	1,100,000	17·8	35·8	6·36
Italy ..	91,280	28,601,600	4,000,000	1,250,000	31·3	43·8	13·7
Switzerland	15,976	3,742,000	3,350,000	650,000	18·5	200·9	40·6
Germany ..	208,800	67,903,400	1,425,000	750,100	52·5	6·8	3·59
Great Britain	88,980	40,831,400	900,000	230,000	25·6	10·2	2·58

*Excluding Alaska (area about half million square miles).

Canada: 2,000,000 sq. miles, is taken as the area treated in the Conservation Commission's estimate of available water power, and the area which we may expect

to see fairly thickly settled during the next few decades. The area of the whole Dominion is 3,729,750 sq. miles.

These developed powers include schemes in course of construction and extensions in view, and refer to the ultimate installed turbine capacity.

In Russia * it is estimated that 20,000,000 h.p. is available, of which 1,000,000 h.p. is developed; in Brazil,† 26,000,000 h.p., with 326,000 h.p. developed; in Iceland,‡ 4,000,000 h.p., of which the development of 1,000,000 h.p. is projected; while in Japan § over 1,000,000 h.p. is developed or in course of construction. If to these be added the powers utilized in such other countries as New Zealand and Tasmania, it appears that the total present development amounts to between 18 and 20 million horse-power. The available resources so far mentioned total slightly over 130 million horse-power. To this is to be added the very considerable but as yet unknown possibilities of the various dependencies of the British Empire. In Australia, Papua, British Guiana, East and South Africa, and in India and Ceylon, these are at present being investigated, and a conservative estimate places their continuous output at between 50 and 70 million horse-power.

In Great Britain the total amount of water power is not large. Scotland offers greater possibilities than any other part of the country. Over a considerable extent of the Highland area the rainfall exceeds 60 in. per annum, and this area is studded with natural lochs which form excellent reservoirs at considerable elevation. Nine of the more promising large power sites have recently been investigated,‖ and offer an aggregate output of some 180,000 continuous horse-power. The largest installation as yet developed in the United Kingdom is at Kinlochleven, which gives a continuous output of close on 30,000 h.p. While in England there are larger rivers than in Scotland, there are fewer natural lakes, and the possibility of water-power development is restricted by the general lack of elevation. The powers are in general small and widely distributed, and must usually be developed without storage by utilizing the natural river flow, as has been done, for example, on the Dee at Chester. There are a few large power sites in the mountainous districts of North Wales. A recent investigation ‖ of a number of typical rivers in England and Wales shows that the available output is approximately 10 h.p. per square mile of catchment area, equivalent to about 580,000 continuous horse-power for both countries. Probably not more than one-half of this is capable of commercial development. It is probable that the total commercially possible output of the United Kingdom is in the neighbourhood of 600,000 h.p., of which slightly over 200,000 h.p. has already been developed.

3. Tidal Power.—The idea of utilizing the rise and fall of the tides

* A recent estimate by the Ministry of Ways of Communication (*Electrical Review*, 22nd February, 1918).

† *The Electrical Trade of Brazil*, J. M. Glen. ‡ *Tidens Tegn*, 13th November, 1918.

§ Japanese Department of Agriculture (*Electrical Review*, 10th January, 1919).

‖ By the Board of Trade Water Power Resources Committee.

for power purposes has long been a favourite one, but up to the present no development of this kind of any size has been carried out. Much attention is at the moment being paid to the possibilities of this source of energy in Great Britain and in France. It is too early as yet to say whether any such schemes as have been suggested are commercially feasible. If not at the moment, there can be little doubt that they will ultimately become so, as the cost of fuel-generated energy becomes greater, and in this case a very large amount of energy will become available.

With a spring tidal range of 20 ft., and a range of one-half this amount at neaps, it is possible to generate, by approved methods of operation, an average daily output of approximately 90,000 h.p. hours per square mile of tidal basin area, or, with storage, to give a continuous 24-hour output of 2200 h.p. Such an estuary as the Severn, where an area of 20 sq. miles could readily be utilized with a spring tidal range of 42 ft., would be capable of an average daily output of 8 million horse-power hours, or of about 60 per cent of this with storage adequate to give a continuous output.

The power which may be developed from a tidal basin of a given area depends on the square of the tidal range, and since the cost per horse-power of the turbines and generating machinery increases rapidly as the working head is diminished, the unit cost of such an installation, except as affected by local circumstances, will be smallest where the tidal range is greatest. For this reason the western and especially the south-western coasts of Great Britain and the western coast of France are particularly well suited for such developments, since the tidal range here is greater than in any part of the world with the possible exceptions of the Bay of Fundy, Hudsons Bay, and the southern extremity of South America.

4. Outlet for Hydro-electric Power.—During the eight years between 1910 and 1919, the hydro-electric horse-power developed and in course of development in Canada has increased by almost 150 per cent, and now shows a total of approximately 2,700,000 h.p. A large number of the plants are designed for the addition of further units when required, and the total ultimate capacity amounts to some 3,385,000 h.p. Of the total power installed 1,757,000 h.p. is in central electric stations. The pulp and paper industry utilizes 473,000 h.p.; lighting absorbs 434,000 h.p.; flour- and grist-mills, 43,000 h.p.; lumber- and saw-mills, 38,000 h.p.; and other manufacturing industries, 173,000 h.p. While, as indicated by these figures, the demand for general industrial purposes is continually growing, the chief outlet for hydro-electric power in the near future is likely to be in connection with electro-chemical processes, and probably railroad electrification. The amount of power already used in electro-chemical processes is very large. Thus the world's production of calcium carbide alone requires some 500,000 e.p.h., and when it is remembered that such products as aluminium, carborundum, chromium, cyanide, caustic soda, chlorates and hypochlorites, magnesium, phosphorus, and silicon are only rendered commercially possible by such processes, it will be realized that the future demand for energy for their manufacture is likely to be

extremely large. Nitrogen fixation is also likely to make great demands. In Norway alone over 400,000 e.p.h. is now utilized for this purpose, and in view of the rapid depletion of the natural nitrate deposits, from which four-fifths of the world's nitrogen consumption has hitherto been supplied, and of the diminution in fertility of most of the great wheat- and cotton-growing areas of the world, the production of artificial fertilizers by one or other system of nitrogen fixation must, in the near future, become a question of the utmost importance.

The electrification of railroads has made relatively rapid strides during recent years. In the United States 440 miles of the Chicago, Milwaukee, and St. Paul Railway, comprising 590 miles of single track, have already been electrified, and contracts for further electrical extensions have been let. On the completion of this work, the total length of electrified track will be about 860 miles. The power for operation is obtained from hydro-electric stations. In France much of the line of the Compagnie du Midi has been electrified with the aid of water power, and the electrification of trunk lines in many other countries is at present under consideration. Such developments will open up a very large field for the utilization of water power where this is available.

The economic development of many of the tropical dependencies of the British Empire, whose latent wealth is practically untapped, is directly interconnected with the development of their water-power resources. Not only would an abundant supply of such power enable railroads to be operated, irrigation schemes to be set on foot, and mineral deposits to be tapped and worked, but it would go far toward solving the labour problem, which promises to be one of some difficulty in the future.

5. Cost of Hydraulic Power.—The cost of hydraulic power is made up mainly of charges against capital, the cost of interest, depreciation, sinking fund charges, taxes, and insurance being always much greater than water charges and costs of operation, maintenance, and supplies. These capital charges vary largely with the local circumstances and physical characteristics of the site. Where the available head is great and the storage provided by a natural lake, or where storage is unnecessary, they may be comparatively small. Where, on the other hand, extensive works are required to store the water and to bring it to the power house, the overall cost of power may be largely in excess of that generated by a steam plant.

A recent analysis of the capital costs of seventy representative hydro-electric stations in Canada, with an aggregate capacity of 746,000 h.p., shows an average constructional cost of £14·2 per installed turbine horse-power. This is exclusive of the cost of water rights, land, transmission, and distribution, and the costs are on a pre-war basis. A similar analysis of the capital costs of 338 installations in Sweden, aggregating 695,000 h.p., shows an average cost of £11·7 per turbine horse-power. If these are grouped according to the size of the installation, they show that the average cost ranges from £27 for installations of less than 200 h.p. to £6·8 for an

installation of 20,000 h.p. Provisional estimates by the Board of Trade Water Power Resources Committee of the cost of development of nine large schemes in Scotland show an average cost, on a pre-war scale, of approximately £25 per installed turbine horse-power.

The total charges may be largely increased by the necessity for long-distance transmission. Thus while the cost of power in bulk at Niagara Falls is 9 dollars per horse-power year, this is increased by transformation and transmission charges to 11·5 dollars at the Falls, to 14 dollars at Hamilton, 54 miles distant, and to 38 dollars at Walkerville, 237 miles distant.

It is evident from these various figures that it is impossible to give any generally applicable figure for the cost of hydro-electric power, or for the limiting cost at which such developments may become economically feasible. This depends essentially on the cost of development and operation of any competing power, and on the purpose for which the output is to be used. A Scottish development, for example, might be economically sound if intended to supply power for general industrial purposes in competition with coal-fired stations, and absolutely unsound if intended to compete for nitrogen fixation or the manufacture of carbide with the extremely cheaply developed water powers of Norway.

6. Necessity for Preliminary Investigation.—In spite of the great importance of water powers, many of the potential powers in existence must of necessity prove economically useless, either on account of their distance from centres of industry, the lack of transport facilities, or from the fact that the storage necessary to give a sufficiently continuous supply would be too costly. Of many potential powers it can be said without further investigation that for the present this is, and for a long time to come will be, the case. Of others the reverse is true. But in the majority of cases the extent to which a scheme is capable of economic development can only be determined after a careful examination of the catchment area and site of the proposed works; after a prolonged investigation of the run-off or rainfall records; and, in an undeveloped country, after an investigation of the mineral and forestal or agricultural possibilities of the surrounding region.

CHAPTER II

Rainfall and Run-off

7. Rainfall.—The mean annual rainfall of the world is about 36 in. per annum, varying from almost zero in its arid regions to between 400 and 500 in. at isolated points in Western Assam. The rainfall of a given district varies greatly with its situation and physical configuration, and with the direction of the prevailing winds. Where these are charged with moisture, through crossing a large stretch of water, the rainfall of

the first high ground encountered by them is always heavy. On the other hand, the rainfall of a district is small if the prevailing winds traverse a large expanse of land before reaching it, or if they come from a district of high to one of lower elevation.

Thus, as indicated by the rainfall map of fig. 1,* the hills on the western coasts of the British Isles precipitate the moisture in the prevailing south-west winds from the Atlantic. The average rainfall at Ben Lomond at an elevation of 1800 ft. is 90·6 in., and at Seathwaite, in Cumberland, is as high as 150 in. The eastern coast of Scotland has a rainfall of about 27 in., and the flat districts east and south-east of the Pennine Chain in England have a rainfall not greatly exceeding 20 in. per annum. The greatest rainfall is usually experienced upon the leeward side of a range of hills, within a few miles of the summit.

8. Fluctuations of Rainfall.—In any district the rainfall varies greatly from month to month throughout the year. The following table indicates typical fluctuations as experienced at Greenwich, in Central Queensland, and at Madras, Bombay, and Cherrapunji, which is situated on the Khasia Hills in south-west Assam at an elevation of 4000 ft., and is reputed to have the heaviest rainfall on record. The hills rise abruptly from the low-lying lands of Caher and Sylhet. At Sylhet, only twenty miles away, the annual rainfall is only one-third that at the higher elevation. The figures from India show the large seasonal variation existing in a region exposed to monsoon winds.

	Monthly Rainfall in Inches												Total Annual.
	J.	F.	M.	A.	M.	J.	J.	A.	S.	O.	N.	D.	
Greenwich (mean of 50 years)	1·99	1·48	1·46	1·66	2·00	2·02	2·47	2·35	2·25	2·88	2·27	1·77	24·6
Central Queensland (mean of 33 years)	7·68	8·47	4·81	2·10	1·01	2·35	1·86	0·82	1·54	1·66	2·35	4·74	40·63
Madras (mean of 33 years)	·91	·33	·18	·61	1·08	1·90	4·09	4·94	5·14	11·27	12·78	6·25	49·5
Bombay (mean of 39 years)	·07	·03	·01	·03	·51	20·88	28·00	15·95	12·50	2·00	·34	·06	80·4
Cherrapunji (mean of 40 years)	·69	2·02	12·12	33·07	43·84	95·72	99·52	76·09	47·41	13·83	1·45	·20	426·0

In addition to this seasonal fluctuation, the rainfall suffers annual fluctuations as indicated by fig. 2, which shows typical records of annual rainfall over a period of thirty years. From these curves it appears, as is generally true, that two, and often three, relatively dry years usually follow each other.

Sir Alexander Binnie,† as a result of an extended investigation, gives

* *Proc. Inst. C. E.*, Vol. CLV, 1903–4, Part I.

† *Proc. Inst. C. E.*, Vol. CIX, 1891–2, Part III. Also *Rainfall, Reservoirs, and Water Supply* (Constable & Co., London, 1913.

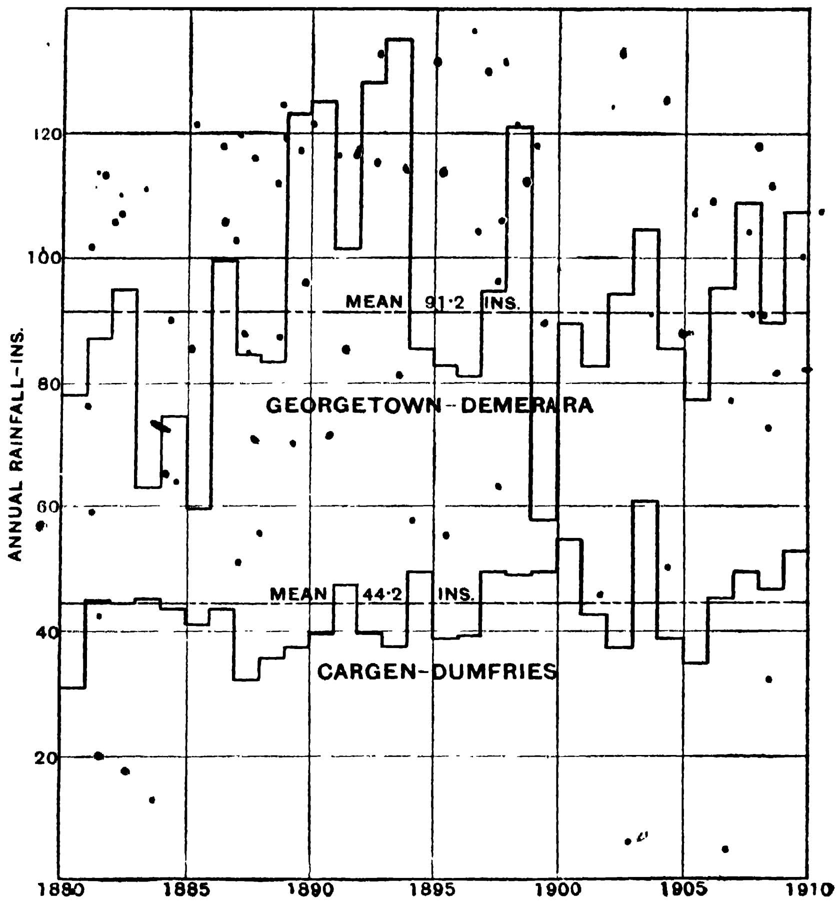


Fig. 2. Rainfall Records at Dumfries and Georgetown, Demerara

the results summarized in table on p. 9, from which the following broad facts emerge:—

1. The wettest year has a rainfall about 50 per cent greater than the average.

2. The driest year has a rainfall about 40 per cent less than the average.

3. The two wettest consecutive years each 35 per cent greater than the average.

4. The three wettest consecutive years each 27 per cent greater than the average.

5. The two driest consecutive years each 30 per cent less than the average.

6. The three driest consecutive years each 25 per cent less than the average.

For the British Isles these figures are (1) 45 per cent; (2) 34 per cent; (3) 30 per cent; (4) 23 per cent; (5) 27 per cent; (6) 22 per cent.

9. Average Rainfall.—A true value of the average rainfall at any particular place can only be obtained as a result of a long extended series of observations. In temperate climates the result of any one year's observations may be 50 per cent above or 40 per cent below the true mean; the mean of five years' observations may be 15 per cent above or below the true mean; of ten years' observations, 8 per cent above or below; and of fifteen years' observations, 5 per cent above or below. In tropical climates the error may be greater. It is only by observations extending over from thirty to thirty-five years that a value within 2 per cent of the mean can be obtained.

FLUCTUATIONS OF RAINFALL. MEAN ANNUAL FALL TAKEN AS UNITY

Stations.	Relative Fall of Wettest and Driest Years.		Average Fall of Two Wettest and Two Driest Consecutive Years.		Average Fall of Three Wettest and Three Driest Consecutive Years.	
	Wettest.	Driest.	Wettest.	Driest.	Wettest.	Driest.
British Isles	1·45	·66	1·30	·73	1·23	·78
Norway, Denmark, Holland, and Belgium ..	1·48	·61	1·33	·66	1·26	·75
France	1·61	·59	1·42	·68	1·31	·74
Italy	1·59	·55	1·39	·70	1·29	·76
Switzerland	1·47	·55	1·35	·62	1·30	·68
Germany	1·39	·61	1·27	·70	1·21	·77
Austria	1·44	·56	1·33	·68	1·27	·76
Russia	1·66	·53	1·46	·63	1·35	·68
India	1·62	·52	1·42	·66	1·30	·72
Africa	1·66	·53	1·51	·64	1·42	·68
Australia	1·56	·53	1·39	·71	1·32	·75
United States and Canada	1·41	·68	1·31	·75	1·25	·79
South America and West Indies	1·51	·55	1·45	·63	1·35	·69
Averages ..	1·51	·60	1·35	·69	1·27	·75

When investigating a hydro-electric scheme the engineer seldom has access to a long series of rainfall observations in the immediate vicinity of the projected catchment area. In such a case the mean annual fall can only be estimated by a comparison of such records as he can obtain in the required locality, with others taken during the same period at some place *similarly situated*, at which a long continuous record has been kept. If, during this period, the corresponding rainfall at the reference station has been, say, 80 per cent of the true mean, the same ratio may be assumed between the observed and mean rainfall at the projected site. In

this way, if all the circumstances be carefully considered, it is possible to obtain a close approximation to the mean rainfall.

10. Rain-gauges should be installed with the mouth of the guage about 1 ft. above the ground-level, and sufficiently remote from any building, tree, or wall which might affect the rainfall in its vicinity. Anything likely to produce wind eddies near a guage is liable to cause erroneous and generally deficient records. If for local reasons it is necessary to raise the level of the gauge, the reading should be modified, since every foot of elevation up to about 10 ft. causes a diminution of approximately 1 per cent in the rainfall recorded.

In order to obtain a reliable mean, rain-gauges should be spread with a fair degree of regularity over a district. In Great Britain one rain-gauge to each 1000 acres of gathering ground is an adequate allowance. In countries possessing topographical features on a larger scale, this number may be reduced considerably. There is always a tendency, which should be guarded against, to place rain-gauges in the most accessible places. This usually means that the higher parts of a gathering ground, over which the rainfall is normally the highest, receives inadequate attention, and the resultant mean tends to be low.

11. Run-off.—Of the rainfall on a given catchment area, a part is evaporated or transpired from the surface of the ground or from the leaves of growing vegetation; another part, large in winter and small in summer, finds its way directly and quickly into the streams; while the larger part sinks into the ground, whence it very gradually seeps into the streams draining the area. It is this underground supply which enables stream flow to be maintained during protracted periods of slight rainfall. A portion, usually very small except in limestone formations, escapes through underground fissures or permeable strata, and appears, if at all, in some other watershed. In the majority of cases it may be assumed that the whole of the rainfall on a given watershed is either evaporated or ultimately appears as stream flow or run-off.

The immediate relationship between rainfall and run-off depends upon a large number of factors, including the geology, topography, and size of the catchment area, the temperature, the vegetation, and the distribution of the rainfall. Where the ground is impervious and steep, a large proportion of the rainfall quickly finds its way into the streams, and, especially on a small catchment area, the run-off more or less closely follows the rainfall. Such an area is very liable to floods and to extremes of drought. On the other hand, if the ground is pervious and flat, and especially if the area is large, it may be weeks before the rain falling on its extreme borders finds its way into the streams. In such an area the extremes of wet and dry weather flow are much less marked than in one, for example, having an impervious clayey subsoil.

The type of rainfall also sensibly affects the relationship. Thus a given amount of precipitation concentrated in a few heavy showers will give a greater run-off than the same amount falling in a continuous but

mild downpour. A short shower on dry ground will give no run-off, while rain falling on frozen ground appears almost wholly as run-off.

Temperature affects the run-off in two ways: by its effect on the total rainfall and evaporation, and by the regulation of flow which it produces by holding back from the streams, in winter, water in the frozen ground and in the form of snow. In such a climate as that of the eastern United States and Canada, or of the northern part of Europe, this effect is very marked. In such climates the winter flow is usually small. The rise of temperature in spring, by melting the snows and releasing the ground

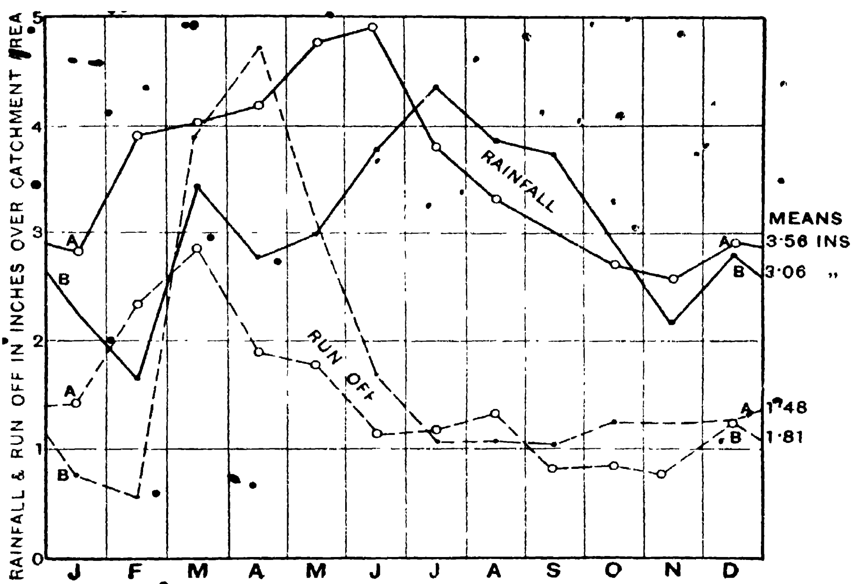


Fig 3.—Rainfall and Run-off Curves for Rivers Roanoke and Connecticut

waters, gives rise to the heaviest flows of the year (fig. 3). Again, in a stream which is fed from a glacier or snowfield at a high elevation, the rising temperature throughout the summer months does much to equalize the summer flow. Such a stream usually undergoes a well-defined change in stage daily, corresponding to the daily fluctuations in temperature.

Vegetation affects the run-off both by its effect on evaporation and on the distribution of flow. It increases the evaporation and hence reduces the total run-off. It increases the ground storage by rendering the ground soil more pervious and absorbent. This effect is relatively greater on a heavy clayey soil than on one which is naturally light and sandy. In cold climates it retards the melting of the snow, especially in forest areas, and thus reduces the tendency to flooding. There is no satisfactory evidence that it has any appreciable effect on the total amount of rainfall. It does,

however, tend to reduce erosion of a river basin, and to reduce the amount of silt carried in suspension.

The discharge from a catchment area thus varies between wide limits, daily, monthly, and yearly, and only bears a very indirect relationship to the rainfall. In this respect each area is a law to itself, and while rainfall records are valuable, they are not in themselves sufficient to enable the probable run-off and its variations to be determined with any accuracy. The exception to this generalization is in the case of an area similarly situated to one of the same general characteristics and geological formation, and of approximately the same size, whose run-off has been gauged, and for which the connection between run-off and rainfall has been obtained. In such a case, the rainfall records on the second area will enable its run-off to be determined with a fair degree of accuracy.

The curves of fig. 3 show the average rainfall and run-off for two rivers of the eastern United States.* The curves AA refer to the Roanoke, with a drainage area of 390 sq. miles, while the curves BB refer to the Connecticut, with an area of 3300 sq. miles.

12. Evaporation.—The ratio of the total run-off to rainfall over an extended period depends almost entirely on the evaporation. Unfortunately the amount of evaporation depends on so many factors, including the mean temperature, the exposure, the type and amount of vegetation and the state of its growth, and the velocity and humidity of the prevailing winds, that only a rough estimate can be made of its value.

The evaporation from a water surface is measured by determining the loss from an evaporation pan, usually about 3 ft. square. This may conveniently rest on floats on the body of water under investigation. Evaporation from land is measured from isolated areas surrounded by deep ditches, where the water draining off is caught and measured. The difference between the measured rainfall and the percolation gives the estimated loss by evaporation and transpiration.

The average evaporation from a free water surface is approximately as follows:—*

Mean temperature, F. ..	30°	40°	50°	60°	70°	80°
Monthly evaporation (inches)	0·5	1·1	1·9	3·5	5·5	8·0

Records of evaporation from such surfaces in the United States show values ranging from 20 to 40 in. per annum in the eastern States, and from 60 to 125 in. in the dry western States.† In Great Britain the value varies from about 10 to 22 in. The figures given in the following table show the mean monthly evaporation from water surfaces at the Derwent Valley ‡

* From reports of U. S. Weather Bureau.

† Parker, *Control of Water* (Routledge), p. 191.

‡ E. Sandeman, M.Inst.C.E., *Proc. Inst. C. E.*, 1912–3.

Water Works in Derbyshire, and at the Talla Reservoir in Dumfries,* in each case for the years 1906 to 1912.

	Jan.	Feb.	Mar.	April.	May.	June.	July.	Aug.	Sept.	Oct.	Nov.	Dec	Total.
Derwent Valley	—	·02	·35	1·08	1·70	1·86	2·37	1·96	1·24	·65	·32	—	11·55
Talla Reservoir	·32	·31	·56	1·05	2·60	2·80	2·90	2·20	1·42	·95	·42	·45	15·64

During these years the annual evaporation at the Derwent ranged from 10·25 to 19·62 in., and at the Talla Reservoir, from 13·00 to 22·91 in.

The evaporation from ground surfaces depends largely on the nature and extent of the vegetation, and also on the amount and distribution of summer rainfall. Experiments show that during the growing season cereal crops absorb from 15 to 25 in. of water, timber from 9 to 12 in., while long grass may absorb as much as 37 in. The average evaporation from a moist bare earth surface is about 60 per cent of that from a water surface, while that from a moist surface covered with short grass is about 90 per cent greater than from a water surface. While little definite information is available, it would appear that in the normal catchment area in temperate climates the total annual evaporation may be taken as equal to that from a free water surface without any very serious error. In hot climates the evaporation from the catchment area as a whole will, in general, be much less than that from a water surface of the same area. In such climates the apparent evaporation from the water surface of a reservoir is often much greater than that deduced from evaporation-pan tests, owing to seepage into the surrounding ground, and to evaporation from the luxuriant growth of vegetation to which this gives rise. This effect is most pronounced in a long and narrow reservoir. In Great Britain experience shows that the mean annual evaporation from the normal moorland catchment area varies from 12 to 18 in. The lower value applies to steep and impervious areas, and the higher to areas having flat and pervious slopes.

In view of the influence of vegetation in affecting the evaporation, the water year may conveniently be divided into two periods, one embracing the six months of more or less vigorous growth when evaporation is large, and the other the remainder of the year, when evaporation is comparatively small.

Vermeule,† as a result of an investigation of a number of catchment areas in the eastern United States, gives the formulæ:

$$E = 15{\cdot}5 + 0{\cdot}16\,R \text{ for the whole year,}$$
$$E = 4{\cdot}2 + 0{\cdot}12\,R \text{ for the period December to May inclusive,}$$
$$E = 11{\cdot}3 + 0{\cdot}20\,R \text{ for the period June to November inclusive,}$$

where E and R are the total evaporation and rainfall in these periods.

* W. R. Reid, *Proc. Inst. C. E.*, 1912-3. † Report of Geol. Survey of New Jersey, 1904.

He states that these figures apply to a climate whose mean annual temperature is approximately 50° F., and that for any other mean temperature they are to be multiplied by (0·05 T° – 1·48), where T° is the mean annual temperature.

A more extended investigation,* however, indicates that while the evaporation does increase with temperature and rainfall, other factors, among which humidity, wind strength, and distribution of summer rainfall

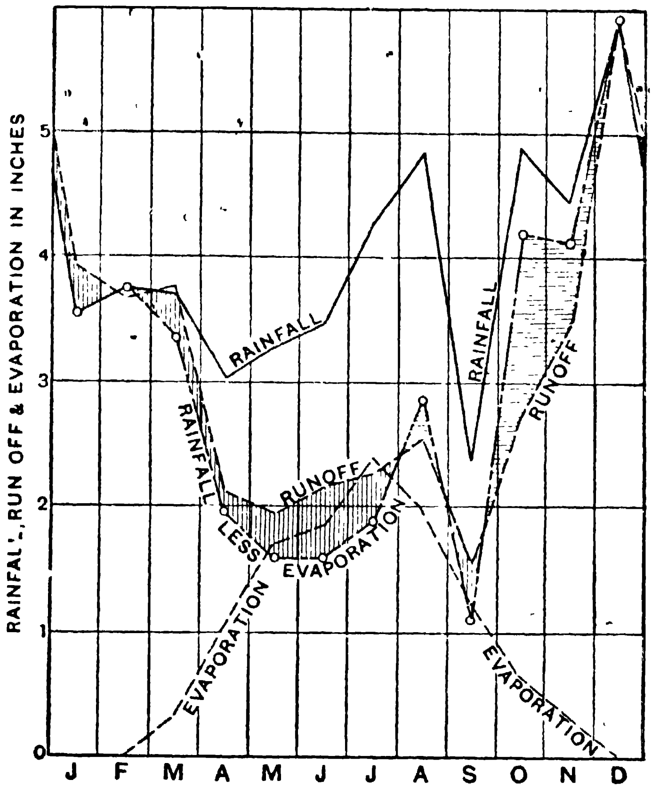


Fig. 4.—Run-off, Rainfall, and Evaporation Measurements on River Derwent Watershed, Derbyshire. Means of readings from 1906 to 1912.

are probably most important, are liable to produce large and irregular departures from any such general law.

13. Ground Storage.—The difference between rainfall and evaporation is available for providing run-off. Owing to the influence of ground storage, the run-off in any given month is, however, not usually equal to the difference between the rainfall and evaporation in that month. This point is illustrated by the curves of fig. 4, which show the rainfall, the evaporation, the difference between rainfall and run-off, and the measured run-off on the watershed of the River Derwent in Derbyshire. The

* *U.S. Water Supply and Irrigation*, Paper No. 80.

values are the means of monthly readings taken over a period from 1906 to 1912*. The catchment area consists of rough moorland with considerable areas of peat, and with steep slopes. The total area is about 50 sq. miles, and the mean annual rainfall over this period is 44·63 in. The elevation of the area ranges from 540 to 2060 ft. The evaporation was measured from a water-tank at an elevation of 800 ft. This evaporation does not necessarily represent that on the catchment area. It has, however, been assumed for purposes of illustration that this is the case. From these curves it appears that from the beginning of the year to the end of July the run-off is greater than the rainfall less evaporation. This indicates

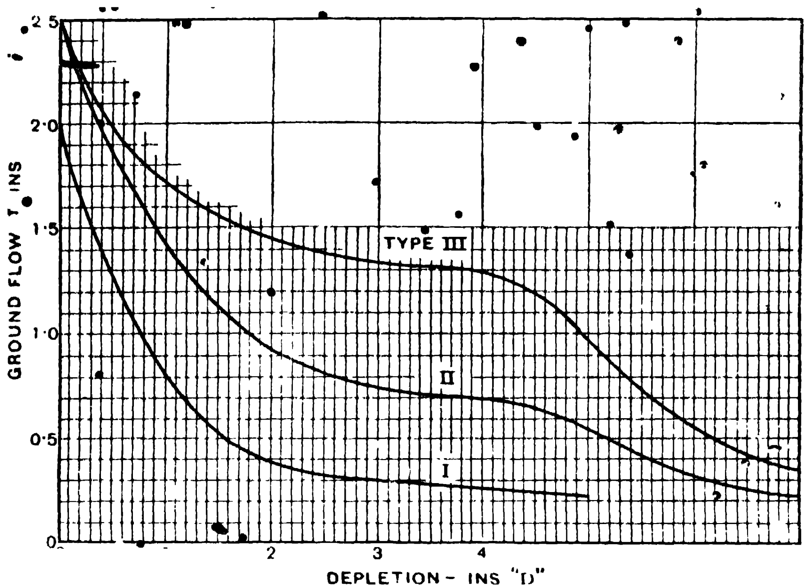


Fig. 5.—Vermeule's Ground Storage Curves

that the ground is giving up stored water as represented by the vertically hatched areas. From September to the middle of December the run-off is less than the rainfall less evaporation, and ground storage is taking place as represented by the horizontally hatched areas.

Vermeule has suggested a general method of deducing the monthly run-off from a knowledge of the monthly rainfall and evaporation by taking into account the effect of ground storage. For this he divides catchment areas into three broad classes:

I. Highland areas of bold relief, and overlying impermeable strata.

II. Areas overlying permeable strata, but with no lakes or swamps.

III. Areas overlying permeable strata with large swamps or surface storage. This covers moorland areas with small lakes and deep accumulations of peat.

* *Proc. Inst. C. E.*, 1912–3.

For each of these Vermeule gives a curve (fig. 5), the ordinates of which represent the monthly run-off in inches, while the abscissæ represent the depletion of the ground storage D, also in inches.

Calling the monthly ground supply f inches, let d_1 be the depletion at the end of the month preceding the one under investigation. Then d_2, the depletion at the end of the month under consideration, will be $d_1 + e + f - r$. The average depletion over the month will be approximately:

$$
\begin{aligned}
\mathrm{D} &= \frac{d_1 + d_2}{2} \\
&= d_1 + \frac{e + f - r}{2} \\
&= d_1 + \frac{f}{2} - \frac{r - e}{2}.
\end{aligned}
$$

The method of utilizing Vermeule's results is best shown by an example, such as the case where the catchment area is of Class II, and where the monthly rainfall and evaporation are as given in the table on p. 17. On the assumption that the ground storage is full at the end of the year, and does not begin to be depleted until the end of March, we have, for the

month of April, 2·08; ·87; = ·61;

$$\mathrm{D} \quad f \quad ·61.$$

From curve II, fig. 5, we find, by trial and error, that $f = 1·99$, $\mathrm{D} = ·38$ satisfies the condition, in that these are the co-ordinates of a point on the curve, and satisfy the above equation. Then, since $d_2 - d_1 = e + f - r$, while $d_1 = 0$, we have

$$
\begin{aligned}
d_2 &= ·87 + 1·99 - 2·08 \\
&= ·78 \text{ in.,}
\end{aligned}
$$

or the ground storage is depleted by ·78 in. during the month. Now, examining the next month, we have $d_1 = ·78$; $r = 2·80$; $e = 1·95$.

$$
\begin{aligned}
\mathrm{D} &= d_1 + \frac{f}{2} - \frac{r - e}{2} \\
&= ·78 + \frac{f}{2} - ·42 \\
&= ·36 + \frac{f}{2}.
\end{aligned}
$$

From the curve we find that $f = 1·40$, $\mathrm{D} = 1·05$ satisfies the conditions. Then again, since $d_2 - d_1 = e + f - r$, we have

$$
\begin{aligned}
d_2 - d_1 &= 1·95 + 1·40 - 2·80 \\
&= 0·55 \text{ in.}
\end{aligned}
$$

as the depletion of ground storage during May, so that at the end of May the total depletion is ·78 + ·55 = 1·33 in. Proceeding in this way from month to month, we get the figures of the following table.

	Rainfall.	Evaporation.	Run-off. r – e.	Run-off. From Ground Storage.	Run-off. Total.	Depletion of Ground Storage.
	inches	inches	inches	inches	inches	inches
Jan.	2·79	·44	2·35	0	2·35	0 (full)
Feb.	2·37	·43	1·04	0	1·94	0 ,,
Mar.	2·04	·54	1·50	0	1·50	0 ,,
April	2·08	·87	1·21	0·78	1·99	·78
May	2·80	1·95	0·85	0·55	1·40	1·33
June	2·84	2·57	0·27	0·76	1·03	2·09
July	3·07	3·14	0·07	0·87	0·80	2·96
Aug.	3·05	2·71	0·34	0·40	0·74	3·36
Sept.	3·15	1·81	1·34	·60	0·74	2·76
Oct.	4·08	1·10	2·98	1·98	1·00	·78
Nov.	3·17	·78	2·39	0·55	1·84	·23
Dec.	2·78	·56	2·22	0·03	2·19	·20
Total,	34·22	16·90			17·52	

In this particular case it will be seen that the ground storage is slightly depleted at the end of the year. The figures show that the ground storage is drawn upon from the beginning of April to the beginning of September, after which it begins to replenish.

The curves of fig. 6 show, for this particular case, the monthly rainfall; the rainfall less evaporation; and the rainfall less evaporation plus the amount drawn from ground storage, which gives the net run-off. It is evident from these curves, that while ground storage does not affect the total run-off over a long interval of time, it does profoundly modify the fluctuations of run-off.

In adopting Vermeule's method, everything, however, depends on the accuracy of the assumptions, and while it has been shown capable of giving good results for areas similar to those from which the original data were obtained, its general application can only follow an extended investigation on other types of watershed. It may be emphasized again at this stage, that measurements of stream flow form the only reliable data on which any close approximation to the water available can be based.

13a. Flood Discharges.—The flood discharge of a river may be some hundred times its average discharge, and since the spillways in the head works must be capable of dealing with the maximum discharge, it is important that some idea of the flood discharge should be obtained. Many efforts have been made to deduce some expression for this, which would be applicable to all streams,* but none of the formulæ suggested

* *Trans. Am. Soc. C. E.*, Vol. LXXVII, pp. 564–694

have proved of general application, owing to the many varying factors involved. Mr. J. T. Whistler,* from an examination of some 600 records of American and European rivers and streams, suggests the formulæ:

$$q_{max.} = 5C\left(\frac{10000}{M + 1000}\right) + 1{\cdot}5C \text{ sec. ft. per square mile,}$$

$$q = 5C\left(\frac{10000}{M + 1000}\right) + 1{\cdot}5C \quad \text{,,} \quad \text{,,} \quad \text{,,}$$

for the maximum flood discharge, and for the mean rate of flood discharge during the worst twenty-four hours. The latter is in general the value to adopt in the design of spillways. Here M is the drainage area in square miles.

The value of C varies from approximately 1·0 to 5·0. Its value in any particular case can only be determined from observations on streams having

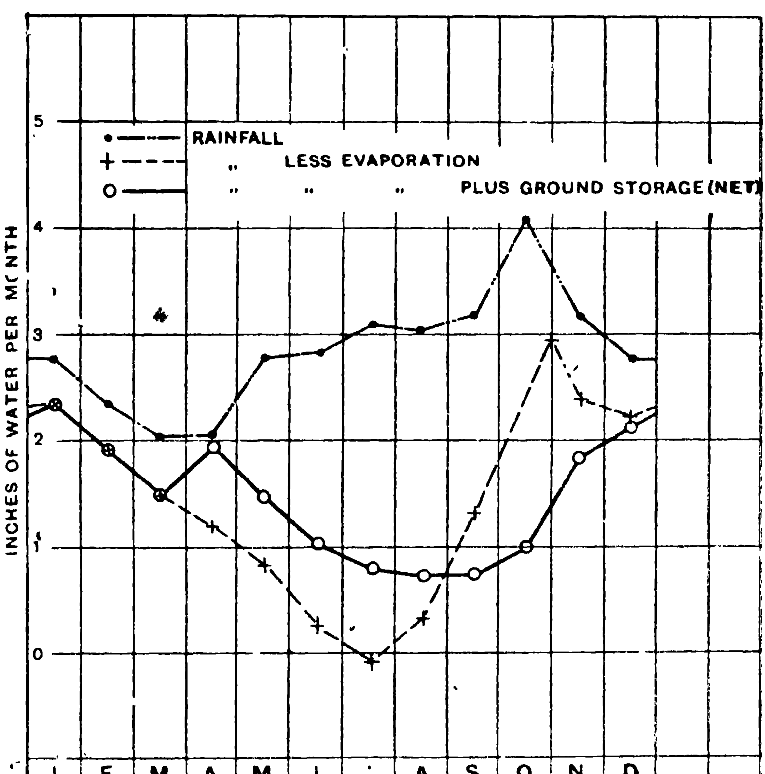


Fig. 6

drainage areas with similar characteristics. The following table * shows the maximum flood discharges for a number of rivers, mainly European:—

* *Engineering News Record*, Vol. LXXXIII, July, 1919, p. 28.

River.	Country.	Dates.	Drainage Area, sq. miles.	Flood Discharge sec. ft. per sq. mile.
Loire	France	?	6945	46
Durance	Bonpas, France	1886	5714	37
Neckar	Germany	?	4770	33
Loire	France	1846	2220	113
Rhine	Germany	?	1620	75
Ombrone	Italy	?	1620	43
Ardèche	France	1827	938	264
Towbrapoorny	India	?	587	324
Kinzig	Germany	?	550	77
Tessin	Bellinzona, Italy	?	541	165
Lausitzer	Germany	1880	481	60
Olsa	Germany	?	435	77
Kinzig	Germany	?	386	100
Iller	Germany	?	367	74
Ubaye	Mouth, France	1843	361	127
Bleone	Mouth, France	1843	351	116
Irrity	India	?	336	450
Ostrawitza	Germany	1894	313	110
Olsa	Germany	?	291	72
Murg	Germany	?	246	100
Serein	France	?	226	78
Cure	France	?	208	77
Bruna	Italy	?	189	189
Elz	Germany	?	185	86
Murg	Germany	?	184	123
Queis	Germany	1888	183·5	164
Ardèche	Aubanas, France	1890	178	694
Wiese	Germany	?	163	108
Brenne	France	1874	145	86
Mandan	Germany	1890	119	125
Wittig	Germany	1880	122	122
Queis	Germany	1888	116	164
Serein	France	?	108·3	98
Stream in	Switzerland	?	87·5	.10
Toss	Germany	1876	63	135
Pliessnitz	Germany	1890	58	820
Mandan	Germany	1887	50·2	210
Zacken	Germany	?	42·5	490
Bargaglino	Genoa, Italy	1892	35·6	485
Eyach	Balingen, Wurtemberg	1895	34·7	356
Allaciente	Italy	?	32·4	260
Torside and Rh'd Reservoir	England	1852	24·1	160
Landwassar	Germany	1887	20·1	365
Medlock	England	1857	18 8	160
Woodhead Reservoir	England	1849	11	320
Brook, near	Dublin, Ireland	1891	10·8	325
Furens	St. Etienne, France	1849	9·6	478
Bargaglino	Above Genoa, Italy	1892	8·8	732
Stream in	Hungary	1875	7·9	400
Kemlitz	Germany	1887	7·4	915
Eyach	Margarethausen, Würtemberg	1895	7·3	789
Nebenwasser	Germany	1887	7·2	490
Spree	Germany	1887	6·7	930
Landwasser	Germany	1887	6·1	830
Kemlitz	Germany	?	5·4	980
Landwasser	Germany	1887	3·8	895
Wittgendorfbach	Germany	1887	3·6	1010
Schopsbach	Germany	1887	3·3	455
Dittelsdorfwasser	Germany	1887	2	1144
Willgendorfbach	Germany	1887	1·3	1115

CHAPTER III

The Flow of Water and its Measurement

Stream flow; gauging of flow; rating tables; gauging stations.

14. Stream Flow.—As indicated in Chapter II, the flow of any natural stream suffers large seasonal variations, while the flow at a given season varies largely from year to year. The variation depends largely on the size and type of watershed, and on its climatic and physical characteristics. Thus the hydrographs of fig. 7* show the variation of flow in the case of the Ebro (north-east Spain), shown in full lines, and of the Rhine (at Basle), dotted lines. The excessive floods in the case of the former river are largely due to extensive deforestation of its watershed.

Even in the case of a stream in a state of normally steady flow, the flow is never perfectly steady and uniform, but takes place in a series of pulsations, whose periodic time may vary from a few seconds to two or three minutes. These pulsations are due to a variety of causes, to eddy formation at the sides and bottom of the channel, to hollows in the bed, and to bends and falls, all of which produce some irregular disturbance of the flow. In addition to these fluctuations, others of longer period are produced by the action of the wind. In a stream fed from a large lake or reservoir, these latter fluctuations may be comparatively large. Thus in a stream whose mean level is constant, the velocity at a point near the surface may vary periodically by some 20 per cent, and at a point near the bottom by as much as 50 per cent in a short interval of time. These fluctuations of velocity are accompanied by fluctuations of surface level, even in a stream in a state of normally steady flow.

15. Gauging of Stream Flow.—The method to be adopted in stream gauging depends on the size of the stream, its state, and on the degree of accuracy required.

Where the installation of a weir capable of taking the whole flow is feasible, this forms the most accurate method. For a stream of medium size the rectangular weir is most suitable. For small flows the triangular notch has advantages. Where the precautions outlined on p. 74 are taken, weir gaugings are not liable to an error of more than 3 to 5 per cent. For large streams the weir becomes too costly as a temporary measuring device, and if no permanent weir is available the only way of obtaining the discharge is to measure the mean velocity of the stream and to multiply this by the cross-sectional area. The mean velocity may be obtained in a number of ways.

(*a*) By current meter.
(*b*) By floats.
(*c*) By colour or chemical methods.

* *Engineering*, 21st September, 1917, p. 293.

(*d*) By measurements of the slope of the stream, and by the use of Kutter's or Chezy's formula (p. 58). This method is unreliable except in the case of an artificial channel of uniform section.

Before considering these methods in detail, a few general observations as to their relative advantages may be made. Owing to the pulsations of velocity existing with normally steady flow, a light float, which tends to record the velocity due to a single pulsation, may give results seriously in error, and in order to obtain a fair estimate of the mean velocity over a given stretch of the stream by means of floats, the average of a large number of observations must be taken. Also, in connection with float measurements, it is necessary to take soundings at a sufficient number of sections to give a true mean of the cross-sectional area of the stream over the whole reach. For these reasons the float method is not well adapted for observations in a channel of irregular section, or in a rising or falling stream.

With current meters, on the other hand, the mean velocity at any point may be obtained with great accuracy if the period of observation is sufficient to cover a series of pulsations of velocity. In general the time of a single observation should not be less than five minutes, and a period of from six to ten minutes is advisable. This method has the advantage that only a single cross section needs to be determined.

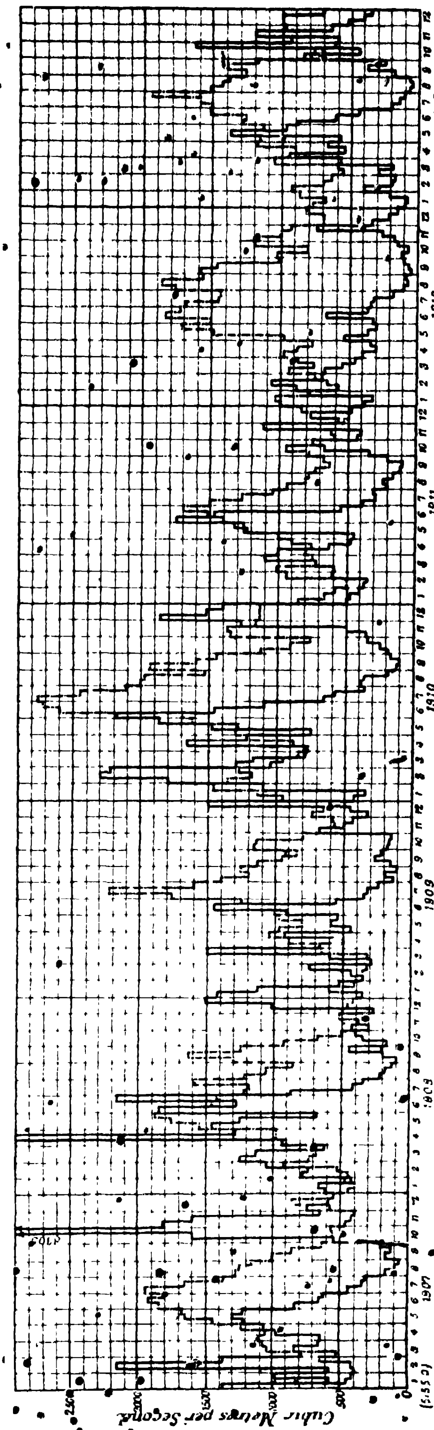


Fig. 7.—Hydrographs of Rivers Ebro and Rhine

By injecting colour into a stream and measuring the time taken by this to traverse a known distance, the mean velocity can be obtained, and if the injection is sufficient to colour an appreciable portion of the cross section, and if the section is regular, this method is capable of giving excellent results. It takes little time, no delicate apparatus, and is much to be preferred to the use of floats.

Whatever method of velocity measurement be adopted, the accuracy of the results depends largely on the physical characteristics of the stream at the point of measurement. This should lie on a straight reach, away from the influence of bends, bridge piers, or dams, and the bed should

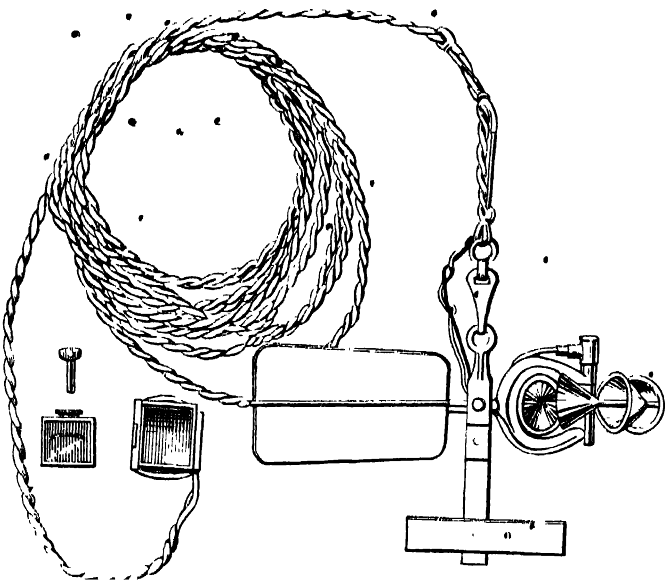


Fig. 8 —Current Meter

be permanent and not strewn with boulders or overgrown with weeds. The banks should be sufficiently high to prevent overflow in times of flood, and the general shape of the cross section such that at all stages of the stream the velocity at all parts of the section may be easily measured.

16. Current Meters.—Various types of current meter are in use. Probably the most generally used is the Price meter (fig. 8). The meter is suspended from a rod or cable, and is provided with a guide vane which keeps its axis perpendicular to the direction of the current. The instrument may be obtained with the wheel geared to a counter which records the revolutions directly, and is put into and out of gear by means of a cord from the point of suspension of the meter, or may make and break the contact in an electrical circuit at each fifth or tenth revolution, thus enabling the number of revolutions to be indicated by means of a buzzer or telephone carried by the observer. The latter type is now almost universal.

The instrument is previously calibrated by towing at known velocities

through still water, the number of revolutions corresponding to these velocities being recorded. It has the disadvantages that it cannot be used where floating grass or weed is prevalent, and that it requires rating at frequent intervals. Further, it is unsuitable for very low velocities. The minimum permissible velocity depends on the type of meter, but in general varies from ·3 to 6 in. per second.

17. Meter Observations.—There are two methods of using the meter, namely, the "point" and the "integration" method. In the point method it is held successively at certain points in a cross section. In a shallow stream this may be done by mounting it on a staff which is carried by an observer in waders, and which is held vertically at the required points with one end resting on the bed of the stream. In deeper streams it is attached to a heavy sinker, and is suspended from a convenient bridge or from a car carried by a cable across the stream, or from an outrigger fixed to an anchored boat where the width precludes this.

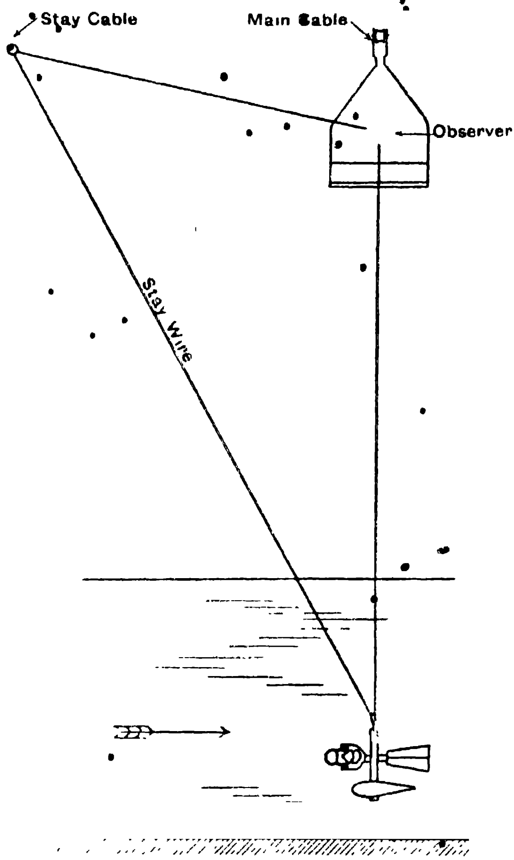


Fig. 9.—Arrangement of Stay Wire at Cable Station

Where the velocities and depths are considerable, a stay line is necessary to hold the meter in the vertical. At a cable station a second light cable is used about 30 ft. up-stream from the measuring section. This carries a light ring through which the stay line is run, as shown in fig. 9. At a bridge station the stay line passes through a pulley carried on an outrigger fixed to the bridge.

When the "point" method is used, the meter may either be held (1) at several equidistant points in a number of equidistant verticals, the mean velocity being deduced from these readings as explained later; (2) at six-tenths, or at mid-depth in a series of equidistant verticals, the mean velocity in each of these verticals then being found by applying a factor; (3) at the surface and bottom only, or at two-tenths and eight-tenths of the depth in a series of verticals; (4) at the surface only. While the first

method gives the most accurate results in a steady stream, the length of time necessary to obtain the many observations is a serious drawback, and renders it unsuitable in a stream which is rising or falling.

In a muddy stream, where it is impossible to see the bottom, owing to the impossibility of using the meter very near to the sides and bottom where the velocities are least, the results tend to be too high. To obviate this the meter should not be placed nearer to the surface than 1 ft.

The depth of the filament of mean velocity in any vertical varies from about ·55*h*. to ·70*h*. In the great majority of cases it lies between ·55*h*. and ·65*h*., increasing with the depth and diminishing as the roughness of the bed increases (p. 64). Generally speaking, the velocity at ·6 of the depth will give the mean velocity in that vertical within 5 per cent, while the velocity at mid-depth multiplied by ·96 will give the mean velocity within about 3 per cent.

Method (3), in which the surface and bottom velocities are measured, is only suitable for very shallow streams. Experiments show that the results are fairly accurate if the bed is smooth or gravelly, the depth from ·4 to 1·0 ft., and the velocity from ·5 to 1·5 ft. per second. With a gravelly bed the meter should be held with its centre from 3 to 4 in. above the bottom and about 2 in. below the surface, while with a smooth bed each distance should be about 2 in. For deeper streams the mean of readings at ·2 h. and ·8 h. is in close agreement with the mean velocity in the vertical, and this method is very often adopted for general stream gauging.

While usually inadvisable to use the surface velocity alone for computing the discharge, it is sometimes impossible in times of flood to make any other measurements. The meter should then be sufficiently submerged to eliminate any disturbance of the surface. Except as affected by the wind, the surface velocity multiplied by a constant which varies from about ·85*h*. in a shallow stream to ·95*h*. in a deep stream (p. 64) gives the mean velocity in a vertical with a fair degree of approximation.

In the "integration" method the meter is kept in motion during its immersion. It should be moved slowly and uniformly from the surface to the bed of the stream and back in a series of verticals. The recorded velocity is then taken as the mean for the particular vertical. Although an observation by this method can be carried out in less time than by the point method, the results are not so accurate. The recorded velocity is the resultant of the velocities of the water and of the meter and, with a meter of the Price type, is higher than the true velocity. The method should only be used where a stream is rising or falling rapidly and where, in consequence, the speed with which the observations can be made is a great advantage.

Soundings.—Simultaneously with the meter observations, soundings should be made from which the cross section of the stream may be obtained. In a narrow stream these should be taken at intervals of from 2 to 5 ft., while where the breadth exceeds 100 ft. the interval should be from 10 to 25 ft., depending on the irregularity of the bed. A graduated

rod is used in depths up to about 15 ft., and a weighted cable in greater depths. With a cable the weight should be lowered until resting on the bottom, and a point marked on the taut cable opposite some fixed point on the observer's platform. The weight is then raised until touching the surface, and the length of cable passing the fixed point in the meantime is measured. In sounding with the rod, care should be taken that this is vertical. Where the bed is soft it is advisable to fix a shoe on the bottom of the rod to prevent it sinking. For a permanent gauging station it is sometimes desirable to grout the bed of the stream for some little distance above and below the section in order to secure permanence of regime.

18. Field Notes.—The following is one method adopted for entering up field observations and computing mean velocities in the case where velocities are measured at several points on a cross section.

Gauging made 14th January, 1914, by X.Y.Z. Meter No. 349 on Esk River. Gauge height: beginning 2·10 ft.; end 2·16 ft.; river rising.

Distance from Initial Point.	Depth of Stream.	Depth of Observations.	Time in Seconds	Total Number of Revolutions	Revolutions per Second	Velocity per Second.	Per Cent of Depth.
feet	feet.	feet					
		0·5	500	600	1·20	2·88	17
		1·0	500	580	1·16	2·78	33
40	3·0	1·5	500	540	1·08	2·59	50
		2·0	500	470	·94	2·27	67
		2·5	500	380	·76	1·83	83

These observations are recorded for a series of verticals in the cross-section. They are then plotted on squared paper, depths as ordinates and velocities as abscissæ, and a smooth curve is drawn through the plotted points, care being taken to give them as nearly as possible equal weight if they do not all fall on a smooth curve. From this curve velocities are read off at top and bottom and at equal intervals of, say, each ·5 ft., and are set down in order. Thus from the above curve we get:

0·0	2·90.	1·5	2·58.
0·5	2·88.	2·0.	2·25.
1·0	2·77.	2·5	1·88.
	3·0	1·31.	

The mean velocity in this vertical is then computed from the prismoidal formula for seven abscissæ as follows:

$$v_m = \tfrac{1}{18}\{v_s + v_b + 4(v_1 + v_3 + v_5) + 2(v_2 + v_4)\}.$$

In this case we have:

$$v_s + v_b = 2{\cdot}90 + 1{\cdot}31 = 4{\cdot}21$$
$$4(v_1 + v_3 + v_5) = 4\{2{\cdot}88 + 2{\cdot}58 + 1{\cdot}88\} = 29{\cdot}36.$$
$$2(v_2 + v_4) = 2\{2{\cdot}77 + 2{\cdot}25\} = 10{\cdot}04.$$
$$\therefore v_m = 2{\cdot}42 \text{ ft. per second.}$$

The cross section having been plotted, the areas of the various compartments having such verticals as their centre lines may be obtained, either by direct measurement, by planimeter, or by calculation, and the discharge calculated as follows:

Compartment.	Area of Section, square feet.	Mean Velocity, feet per second.	Discharge, c.f.s.
1	15·1	1·32	19·9
2	28·2	1·97	55·6
3	36·5	2·42	88·3
4	32·1	2·56	82·1
5	23·7	1·99	47·2
6	13·5	1·33	17·9
			Total 311·0 c.f.s.

When the vertical velocity-curves have been obtained, the discharge may be computed more accurately by considering the discharge between

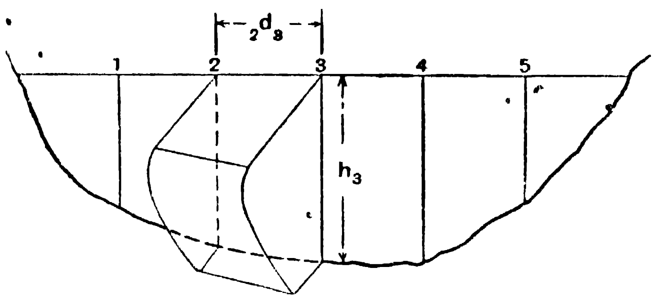


Fig. 10

any two such verticals as being represented by the volume of the solid having these curves bounding opposite parallel sides, as shown in fig. 10. For example, the discharge between the verticals 2 and 3 in this figure is given by

$$\frac{{}_2d_3}{3}\{v_2h_2 + v_3h_3 + \sqrt{v_2h_2v_3h_3}\} \text{ c.f.s.},$$

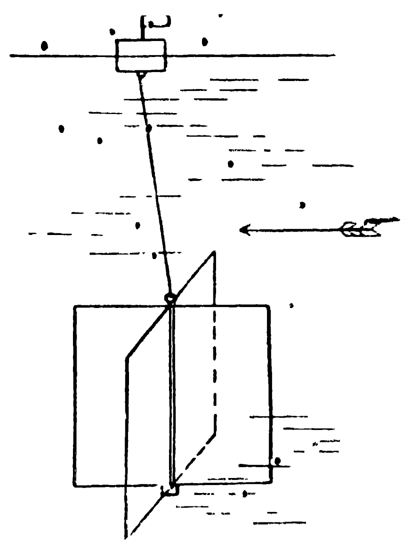
where v_2 and v_3 are mean velocities in the verticals 2 and 3, and where h_2 and h_3 are the corresponding soundings, $_2d_3$ being the distance between the verticals. The discharge between the two end soundings is then given by the sum of such terms as the above between these soundings. To this

must be added the discharge over those sections outside the end soundings, which is given by

$$\tfrac{1}{3}\{v_1 h_1 \times {}_0d_1 + v_5 h_5 \times {}_5d_e\} \text{ c.f.s.}$$

19. Float Measurements.—Floats may be divided into three classes: (1) surface floats; (2) sub-surface floats; (3) rod floats.

Surface floats consist of any easily-seen light floating bodies of small size, so as to move with and register the velocity of the surface filaments. These are liberated at a series of points across the stream at the head of a long straight reach, whose length should be not less than about 200 ft., and the time occupied in covering a measured distance is noted. The surface velocity in each of a number of vertical sections is obtained by repeated observations, and the mean velocity in each vertical is then obtained by multiplying the surface velocity by a factor varying from ·85 to ·95, depending on the depth and condition of the channel (p. 64). The stream sections may be marked, in a channel of moderate width, by ropes hanging from a bridge or temporary support and trailing in the stream. In a large river this method is impracticable, and observations with the theodolite are necessary to determine the path of the float.



Sub-surfa

The effect of the wind on the surface velocity, together with the tendency of the floats to follow every cross current and to be deflected by every surface eddy, renders this method of measurement very unsatisfactory.

Sub-surface floats consist of bodies having surfaces of large area, as illustrated, for example, in fig. 11, attached to small surface floats for ease of observation, the length of connection being adjusted so as to allow the true float to remain at any given depth. The velocity of the float will then be approximately that of the current at the required depth. A series of such floats liberated at different points in the cross section of a stream and at different depths may be used to give, by their mean velocity, the mean of that of the stream; or by arranging a single row, the depth of each being ·6 that of the stream at the point of introduction, these may be taken as giving the mean velocities in their respective sections. While this type is more reliable than the surface float, it suffers from the disadvantage that it is impossible to determine the exact position or depth of the lower float, for while the position of the upper float may be known, that of the lower float varies with the direction and velocity of the wind, and with the length

of connecting cord. Also, the upper float drags the lower whenever the latter is in a region of lower velocity, and since this is the case over the greater part of the depth, it tends to make the velocities of flow recorded by the float too high. Experiments show that the errors involved by the use of such floats may be between 5 per cent and 25 per cent.*

The *rod float* consists of a light wooden rod or tin tube about 1 in. in diameter, and made in adjustable lengths. The lower end of the bottom section is weighted, and the length adjusted until the rod floats vertically, with its lower end clearing the bottom by a few inches. In a large river where these are not likely to interfere with navigation, logs of wood about 12 in. in diameter, having their lower ends weighted with iron and their upper ends painted white, may be used.

The velocity of the rod is approximately the same as the mean over its depth, and gives the mean velocity over the vertical in which it floats. The difficulty in using the rod lies in its tendency to drag over shoals and weeds, and to obviate this its lower end may be arranged to float at a height h_1 above the bed of the stream.

For such a case Francis gives the empirical formula

$$v_m = v_r\left(1{\cdot}012 - {\cdot}116\sqrt{\frac{h_1}{h}}\right)$$

giving the mean velocity in the vertical containing the rod in terms of the velocity of the rod v_r, h_1, and h the depth of the stream. Here h_1 should be less than $\cdot 25h$.

In channels of moderate and uniform depth, the rod float is capable of giving results in close agreement with weir gaugings.†

20. Measurement of Velocity by Colour Injection.—The velocity may be determined by injecting colouring matter into the stream, and noting the time this takes to traverse a measured distance. For successful results the colour must be injected in a single burst. The varying velocities across any section, and the mixing of the filaments, cause the length of the colour band parallel to the axis of the stream to increase gradually with the distance down-stream from the point of injection. At the down-stream measuring point, the times at which the first and last traces of colour pass, and also the time at which the point of maximum colour passes, should be noted. The mean of the first two should agree with the third, and gives the time required. In clear water a solution of permanganate of potash may be used. In waters discoloured by organic matter or vegetable stains, red or green aniline dye gives good results. The method is well adapted to the measurement of the flow in a long penstock. It can be used in a power plant without any interruption to the ordinary load, and requires no special appliances except for injecting the dye near

* *U. S. Water Supply and Irrigation*, Paper No 95, pp. 48 and 49.

† *U. S. Water Supply and Irrigation*, Paper No 95, p 55.

the centre of the penstock. Tests show that the method is capable of giving results in close agreement with weir measurements.*

21. Gauging by Chemical Methods.—By adding a strong solution of some chemical, for which sensitive reagents are available, at a uniform and known rate into a stream, and by collecting and analysing a sample taken from the stream at some point below, where admixture is complete, the volume of flow can readily be computed.

The method is best adapted to rapid and irregular streams in which the admixture is most thorough, and which, incidentally, are most difficult to gauge by other means. The solution should be applied at a number of points in a cross section if the stream is wide, and preferably near mid-depth. Sufficient of the solution should be used to give a uniform supply for about thirty minutes. Various chemicals may be used. Unless the water is brackish, ordinary salt is suitable; if brackish or salty, sulphuric acid or caustic soda give good results. With a saturated solution of common salt a dilution of 1 in 750,000 will give, with silver nitrate, a precipitate weighing 1 mgm. per litre of the sample, and the gravimetric analysis of such a sample will enable an accuracy of 1 per cent to be attained. With sulphuric acid, S.G. 1·838, the dilution for the same degree of accuracy and the same size of sample is 1 in 4,700,000. A very approximate idea of the discharge thus enables the necessary volume of solution for any given period to be calculated.

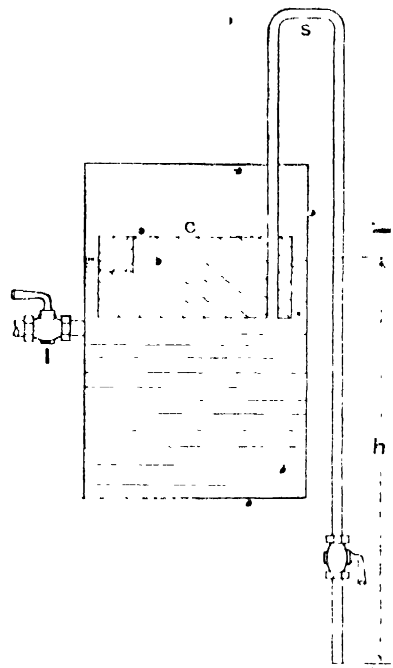


Fig. 12.—Constant Flow Tank for Chemical Gauging

The sample should be taken over a number of points in the cross section. It may be taken by a pump through a flexible pipe whose open end is traversed across the stream during the process. A surface float will give the approximate time necessary for the first of the mixture to reach the sampling section. A few minutes should then be allowed before beginning to take the samples. Immediately before and after the tests, a sample of the stream water should be taken and tested for salinity or acidity, and the results of the mixture tests should be corrected for the pressure of any natural saltness or acidity if present.

* For a description of the details of such tests, see *Engineering News*, 23rd September, 1915, p. 617 (R. Taylor). The dye may readily be introduced in a glass flask which is suspended from a string, and which is broken by a weight sliding down the string.

Fig. 12 * shows one method of introducing the solution at a known and regular rate. A tank, having an internal inlet pipe I is filled with the solution, on which a cork float C, fitted with a siphon pipe S and a balance weight, is floated. Once flow is started the head *h*, and therefore the discharge, is constant. Additional solution is added as required through the pipe I. By shutting off the supply and noting the rate at which the surface-level falls, the rate of discharge can be accurately determined.

22. Gauging of Ice-covered Streams. -When a stream is ice-bound, the flow becomes somewhat similar to that in a closed flume, and may be determined from current-meter observations carried out through holes cut in the ice. Gaugings of ice-bound streams by members of the United States Geological Survey † lead to the following conclusions:

(1) The maximum velocity occurs at a point between ·35 and ·40 of the depth measured from the under side of the ice. The ratio of mean to maximum velocity ranges from about ·80 with a depth of 3 ft. to ·92 with a depth of 16 ft.

(2) There are two points of mean velocity on a vertical, the first lying between ·08 and ·013 of the depth, and the second between ·68 and ·74 of the depth.

(3) In gauging such streams, the vertical velocity curve method or the integration method should be used in preference to any of the single-point and coefficient methods.

23. Stream Rating Curves and Tables. From a series of stream gaugings at different stages of flow, a rating curve or table can be prepared, showing the relationship between the height of water referred to some permanent datum level, and the discharge of the stream. Such a curve is shown in fig. 13. If the channel is permanent, the relationship between height and discharge under conditions of steady flow is fixed, and a continuous record of gauge heights at the gauging station enables a continuous record of the discharge to be obtained.

In preparing the rating curve, the observed discharges are plotted against the gauge heights, and a smooth curve is drawn through the plotted points. In case any of the plotted points are seriously off this curve, it is always advisable to plot also a height-area curve showing the cross-sectional area for each gauge height, and a height-velocity curve showing the measured mean velocity for each height, as in fig. 13. The product of corresponding abscissæ of these curves should equal the abscissæ of the discharge curve, and this often enables an error to be located as due either to an obvious error in the measurement of the area or the velocity at some particular stage. When required to extend a rating curve for higher or lower stages than are covered by the experimental measurements, it is

* C. E. Stromeyer, *Proc. Inst. C. E.*, Vol. CLX, 1904 5, Part II, p. 349. This paper gives the details of various gaugings by this method, and shows that it is capable of giving excellent results.

† *U. S. Water Supply and Irrigation*, Paper No. 95, p. 159.

more accurate to do this by extending the area and mean velocity curves, and to take their product for the extended rating curve.

The rating curve is only accurate for the conditions under which it was obtained. Even with a permanent channel, the discharge for a given gauge height is greater with a rising stream and less with a falling stream than when the flow is steady, owing to the changes in the surface slope and velocity under these conditions. These effects are, however, not important except in extreme cases. Any change in channel, due to erosion or silting, or to the growth or removal of weeds, necessitates the construction

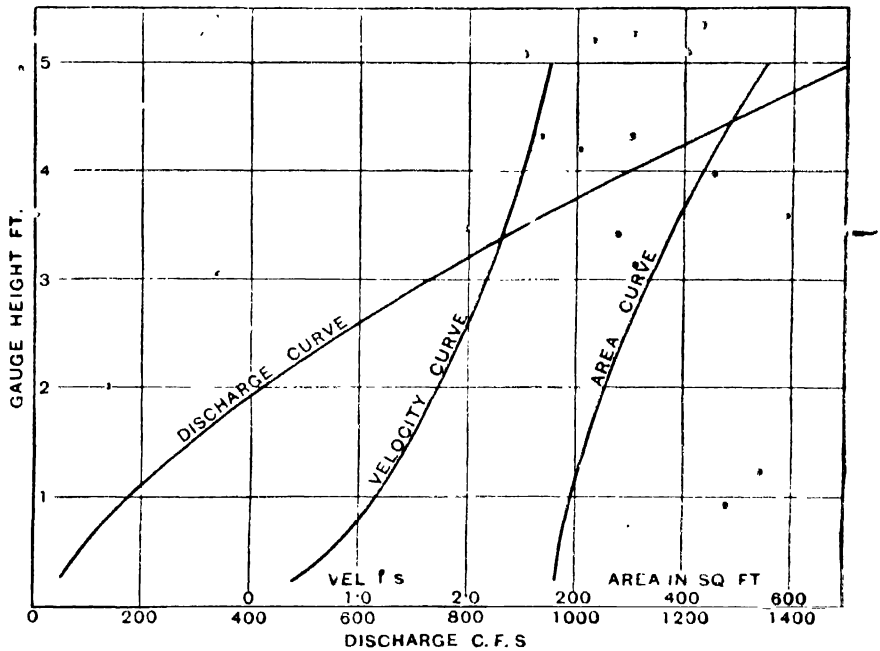


Fig. 13.—Velocity, Area, and Discharge Curves f

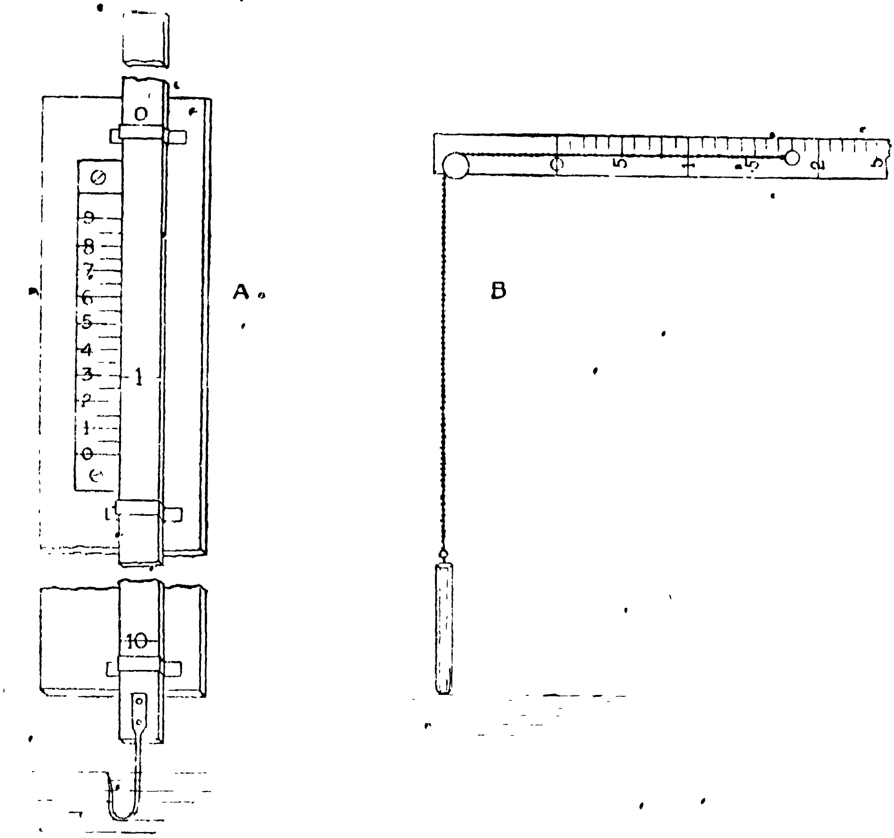
of a fresh curve. The length of time during which a curve may be valid thus depends entirely on the local circumstances, and may range from a few days to several months.

24. Gauging Stations. On a stream subject to changes of channel, or which offers no suitable section for making measurements, it is necessary to construct an artificial control below the gauging station. Such a control may consist of a low dam, whose height is sufficient to prevent its being drowned at high stages by any banking up of the down-stream flow. With such a control the relationship between gauge height and discharge becomes stable, and independent of any change of section at the gauging station itself.

The gauging station is equipped with some form of height gauge, which may record heights automatically, or may require to be read daily.

Automatic gauges are operated by a float, which is situated in a chamber communicating with the river by a small pipe or orifice so as to damp out any wave effects. The float operates the pencil mechanism, which makes a continuous height record on a drum rotated by clockwork.

Non-recording gauges consist of direct gauges, which are fixed graduated staffs or boards on which the water-level is observed directly, or indirect gauges, in which the water-level is transferred to a graduated scale fixed above the water surface.



Hook Gauge Weight Gauge

Fig. 14

In the hook gauge, fig. 14A, a vertical rod graduated in feet carries a hook whose point is adjusted until in the surface. The height is then recorded against the fixed scale, which is usually graduated in tenths of a foot.

The weight gauge, fig. 14B, consists of a graduated horizontal scale board carrying a vertical pulley over which works a light flexible chain carrying a weight and a marker. To use the gauge, the weight is lowered until touching the water, when the height is read off from the graduated scale opposite the marker. These types of gauge have the advantage over

the direct gauge that they are free from danger of displacement from ice and floating debris.

The float gauge consists of a float which carries either a vertical rod or a chain kept taut by a small balance weight. The rod or chain carries a marker which moves over a graduated scale.

In order to enable accurate readings to be obtained with a vertical staff gauge or a float gauge, this should be installed in a stilling box, erected in or alongside the stream, and communicating with the stream through a series of small holes. The zero of the gauge should always be below the lowest water-level.

CHAPTER IV

The Available Power

Available power from a stream; storage; storage calculations; the mass curve; analytical methods; auxiliary and reserve plant; load curves.

25. Available Power. If a development is projected on a stream having no artificial storage, the maximum power which can be developed continuously depends entirely on the water available during the period of minimum flow. In this connection it is usual to define the minimum flow as the mean of the average flows during the lowest two consecutive seven-day periods during the period of record, which should, if possible, extend over at least seven years. This is evidently higher than the absolute minimum on the day of lowest flow, but this absolute minimum does not usually equal the practical minimum of plant capacity which may profitably be installed.

As extreme low-water conditions usually last for only a small portion of the year, it is possible to develop, during the greater part of the year, a much larger amount of power than the minimum. Broadly speaking, an installation laid out to absorb from three to four times the minimum stream-flow can, in temperate climates, be operated at full load for at least half the year, and at an average of about 80 per cent of full load for the remainder of the year. This secondary power may often profitably be utilized in industries which do not employ a large amount of labour, and do not involve a large capital outlay. To determine the maximum practicable economical development, the Water Power Branch of the Department of the Interior, Canada, suggests the following procedure. "The months of each year are arranged according to the day of the lowest flow in each. The lowest of the six high months is taken as the basic month. The average flow of the lowest seven consecutive days in that month determines the maximum available flow for the year. The average of such maximum figures for all years in the period for which data are available is the assumed maximum to be used in calculations." Much,

however, depends on local circumstances. In some cases it is economical to develop up to that amount which can be maintained continuously during the highest four months of the year, the deficiency in power during the remainder of the year being provided by the installation of fuel-power plants as auxiliaries.

Where the installation is intended to supply industries having heavy labour costs, or large capital outlays, and especially in the case of a public supply for lighting and traction, a certain fixed minimum output must be available at all times. In such a case, if the minimum capacity of the stream is not sufficient for the power requirements, this must be augmented either by an auxiliary plant, or by storage of water for use at periods of low flow, or by a combination of the two.

Storage and Pondage.—The term "storage" is usually applied to large impounding schemes designed to give more or less uniform supply during the dry seasons, while the term "pondage" is applied to smaller schemes in which the night flow is impounded for use during the working hours of the day. Where sufficient pondage is available to store the whole flow of a stream, the available horse-power is greatly increased. Thus in a plant operating over an eight-hour day, the storage of the flow during the sixteen idle hours enables the output to be increased to three times the value possible without such storage.

Where the storage of this water leads to a considerable increase in the water-level behind the dam, the effect of this on other power plants farther up the stream, and the effect of the variation in head during the working period, require to be carefully considered.

26. Storage Diagrams.—Where it is required to equalize the flow of a stream over a long period, the necessary reservoir capacity may be determined graphically by drawing a "mass curve". For this the net monthly run-offs, corrected for seepage and evaporation, are added, and plotted as ordinates on a base representing times, so that the ordinate corresponding to any given date represents the total discharge since the commencement of the period. If a tangent be drawn at any point of the curve thus formed, its slope represents the rate of flow at that instant.

In a similar manner the curve of demand can be drawn. If uniform power is to be developed at all seasons, this will be a straight line. If not, if the seasonal demand can be predetermined, the consumption curve can be modified to suit.

Any convenient unit of capacity may be used for the vertical scale. The acre foot, equivalent to 43,560 c. ft., or the million or billion cubic feet are convenient units, depending on the size of the scheme.

If, over any period, the slope of the mass supply curve is greater than that of the demand curve, the reservoir, if not already full, is storing water. Conversely, if the slope of the supply curve is less than that of the demand curve, the reservoir level is falling, while, where the curves are parallel, the rates of inflow and outflow are equal. The vertical intercepts between the supply curve OG and any such demand curve as OL (fig. 15) represent

the total excess of supply over demand from the beginning of the period.

To determine the storage capacity necessary to balance a given demand, represented by the demand curve OL, draw tangents parallel to OL through the various crests of the mass curve, as at BM. The vertical intercept CC′, between this tangent and the supply curve at any point C, represents the excess of outflow over inflow during the period BC, and the greatest of such intercepts over the whole period under consideration

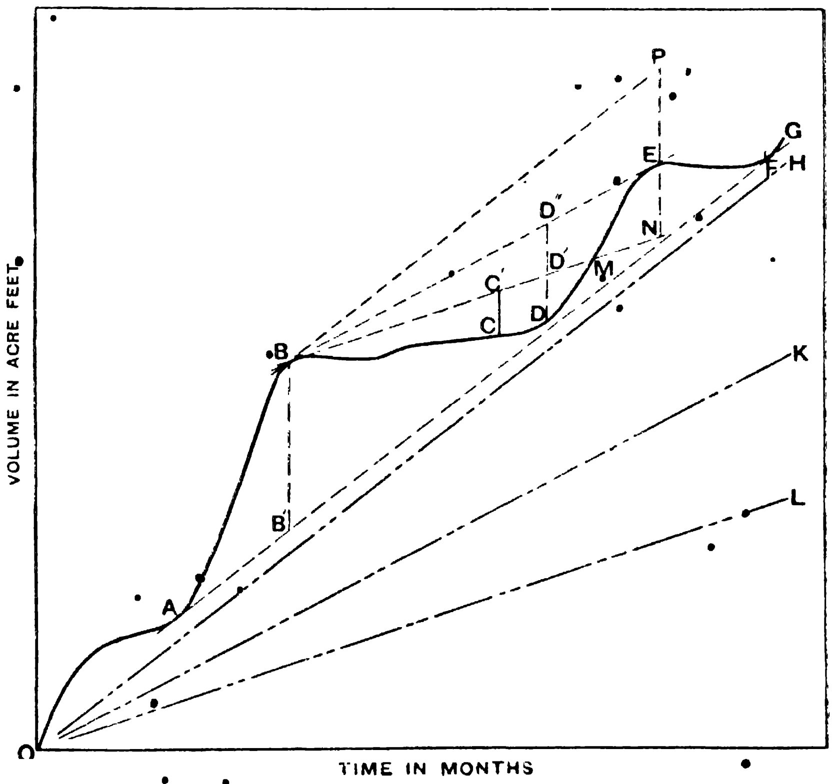


Fig. 15.—Mass Curve for Storage Determination

represents, on the vertical scale, the necessary capacity of the reservoir. In the case in question the maximum intercept occurs about D, and the reservoir, if full at B, and if of capacity DD′, will become empty at D. It will refill at M, and will overflow from M to E, the volume of overflow being represented by NE.

If the tangent does not again intercept the curve, as would be the case with the tangent BP, corresponding to a demand curve OH, the reservoir will not refill within the period under review. The tangent BE, corresponding to the demand curve OK, represents the maximum rate of demand for the reservoir to refill, and with this rate of demand the necessary storage volume is given by the intercept DD″.

In order to determine the maximum demand which can be satisfied by storage, tangents should also be drawn to the troughs of the mass curve as shown at AF. If the reservoir be empty at A, and if AF be the lowest of all such double tangents passing through A, the reservoir will again be empty at F, and the slope of this tangent, which, in the sketch, corresponds to the demand curve OH, gives the maximum possible demand which can be satisfied over this period. The maximum intercept BB′ gives the necessary storage volume. In this case the reservoir would be full at B and almost empty at D, but would be below high-water level by an amount represented by the intercept PE at E.

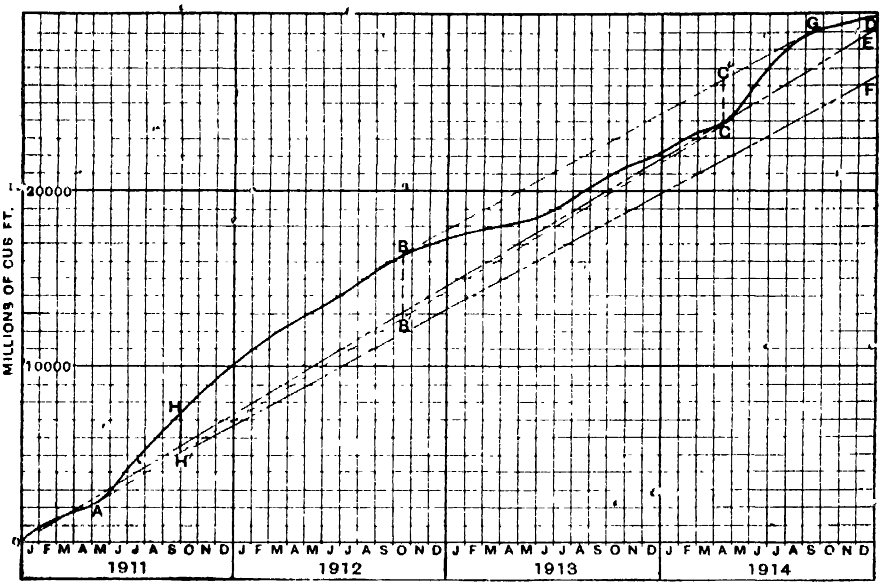


Fig. 16. Typical Mass Curve for River

The mass curve should be drawn to cover as long a period as possible. In regions having a moderately high rainfall a five-year period usually gives results within 15 per cent, although an occasional error as high as 30 per cent may be experienced. A ten-year period may usually be relied upon as being within 10 per cent. In dry regions longer periods are desirable, and even then the results should be used with caution.

Fig. 16 shows a typical mass curve for a river, over a period of four years. Here the ordinates represent millions of cubic feet. The total run-off over the four years is 30,000 million cubic feet, corresponding to an average run-off of 238 c. ft. per second. The storage represented by CC′ equals 2300 million cubic feet, and would enable the demand curve OF, representing 218 c.f.s., to be satisfied. The reservoir would then be full at B and G, but would overflow at H where HH′ is equal to CC′. The storage represented by BB′ equals 4200 million cubic feet, and would enable

the demand curve OE, representing 232 f.s., to be satisfied. The reservoir would then be empty at A and C. It would be full at B, but would be 1900 million cubic feet below high-water level at G. In this case an increase of 83 per cent in storage capacity is only accompanied by an increase of 6·5 per cent in continuous output.

An examination of the effect of various amounts of storage in this particular case shows that the water continuously available at the turbines is given by the ordinates of the curve of fig. 17, which also represents to scale the possible continuous horse-power output of the plant. This diagram strikingly illustrates how the value per million cubic feet of storage capacity becomes relatively less as the storage volume is increased. This is a general characteristic of all storage schemes. In the case in question

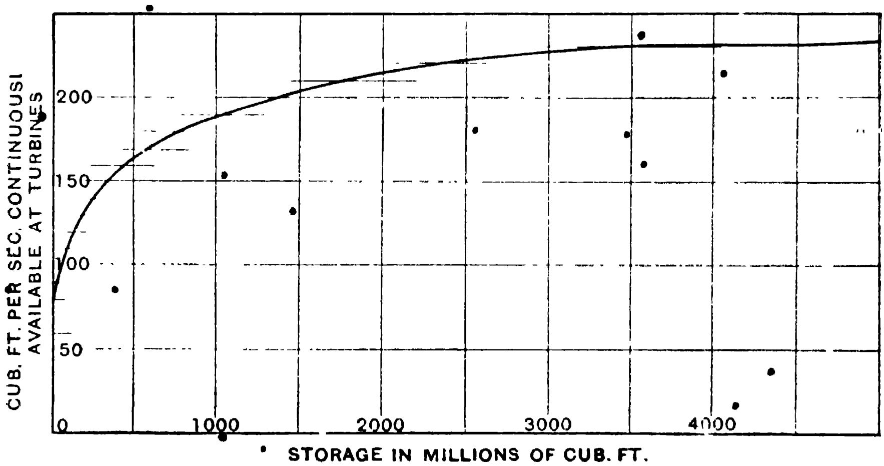


Fig. 17

a storage capacity of 1000 million cubic feet, equivalent to approximately 50 days' supply, enables a continuous flow of 190 c.f.s. to be utilized, or 80 per cent of the maximum possible continuous flow. The output with a 100-days' storage would only be 12½ per cent greater, and with a 150-days' storage 20 per cent greater. No further increase of output is possible by increasing the storage capacity beyond 4200 million cubic feet, which corresponds to 205 days' supply.

The most economical development of a given scheme can only be determined by obtaining in this way the power rendered available by different amounts of storage, and by balancing the economic value of this power against the charges to be debited against the extra storage. Since the power developed by the stored water, per acre of storage area, is proportional to the working head, the cost per horse-power of storage capacity increases rapidly as the head diminishes. The cost of any large storage for a low-head installation is usually prohibitive, and it is only in high-head plant that any large amount of storage will prove to be economical. An

investigation of any particular scheme over a lengthy period will generally show that the full storage, if developed, would only be fully utilized at intervals of from five to ten years. The most economical development will in general assure that the entire volume of stored water can be used practically every year.

In the case in question, if the working head is 250 ft., the continuous horse-power available with 50 days' storage, assuming a turbine efficiency of 80 per cent, would be

$$\frac{190 \times 62{\cdot}4 \times 250 \times {\cdot}8}{550} = 4300 \text{ h.p.},$$

while with 100 days' storage it would be 4840 h.p., and, with 150 days' storage, 5150 h.p. Assuming a depth of storage of 15 ft., a storage area of 1500 acres would be required for 50 days' storage, and 3000 acres and 4500 acres respectively for 100 and 150 days' storage.

At 6 per cent on £20 per acre the annual charge due to the cost of the storage ground alone would be £1800 greater for 50 than for 100 days' storage, and £1800 greater for 100 than for 150 days' storage. Thus the annual cost of the additional power due to this charge alone would be £3·34 per horse-power as between 50 and 100 days' storage, and £5·80 per horse-power as between 100 and 150 days' storage. To this is to be added, of course, the charge due to the extra length and height of dam required and to the extra cost of machinery, power house, pipe line, and transmission.

27. The question of storage may also be investigated by analytical methods, as indicated in the following table, which makes use of the data from which the mass curve of fig. 16 was prepared. From the run-off records, the mean monthly run-off in cubic feet per second and the total monthly run-off are tabulated for each month. It is assumed that the continuous flow required at the turbines is 218 c.f.s., corresponding to a monthly demand of 574 million cubic feet. With the reservoir empty at the end of April, 1911, and with an assumed capacity of 2300 million cubic feet, the volume stored up to the end of each month and the amount wasted when the reservoir is full are obtained step by step, as indicated in the table. With any other storage volume, the demand from the turbines would require to be different for the conditions to balance as in this example, the correct value being readily obtained by trial.

Fig. 18 illustrates a second graphical method of investigating storage conditions on a stream. The diagram represents the hydrograph or discharge curve plotted from the data of the preceding table, and refers to the stream already considered by the mass curve method. Each small square of the diagram represents a flow of 10 c.f.s. over one twenty-fourth of a year, and thus a volume of $365 \times 3600 \times 10 \div 24 = 13{\cdot}15$ million cubic feet.

The horizontal line EE' corresponds to a constant demand of 232 c.f.s., in which case the area from A to C above the line is balanced by the

Year	Date.	Net run-off. c.f.s.	Net run-off. Million c.f. per Month.	Millions of c.f. per Month. To Turbines.	To Storage.	From Storage.	Total Volume in Reservoir.	Volume Wasted. Per Month.	Total.
1911.	Jan.	268	705	574	131	—		—	—
	Feb.	203	534	,,	—	40		—	—
	Mar.	144	379	,,	—	195	—	—	—
	Apr.	131	344	,,	—	230	Empty		—
	May	300	790	,,	216	—	216		—
	June	411	1080	,,	506	—	722	—	
	July	395	1040	,,	466		1188		—
	Aug.	406	1070	,,	496		1684		
	Sept.	451	1190	,,	616		2300 (full)	—	—
	Oct.	426	1170	,,			Full	596	596
	Nov.	395	1090	,,	—	—	,,	516	1112
	Dec.	331	870	,,		—	,,	296	1408
1912.	Jan.	300	790	,,			,,	216	1624
	Feb.	306	805	,,	—	—	,,	231	1855
	Mar.	222	585	,,	—		,,	110	1965
	Apr.	170	448	,,	—	126	2144		—
	May	180	475	,,	—	99	2045	—	—
	June	230	605	,,	31		2076		—
	July	266	700	,,	126		2202		—
	Aug.	242	635	,,	61	—	2263	—	—
	Sept.	232	611	,,	37		2300 (full)		—
	Oct.	180	474	,,		100	2170	—	
	Nov.	136	358	,,		216	1954		—
	Dec.	120	316	,,		258	1696		
1913.	Jan.	102	268	,,		306	1390		
	Feb.	87	228	,,		346	1044	—	
	Mar.	66	174	,,	—	400	644	—	
	Apr.	87	228	,,		346	298		—
	May	118	310	,,	—	264	34		—
	June	236	620	,,	46	—	80		
	July	272	715	,,	141	—	221	—	—
	Aug.	247	650	,,	76		297	—	
	Sept.	236	620	,,	46	—	343		
	Oct.	245	645	,,	71		414		—
	Nov.	186	489	,,		85	329		
	Dec.	150	394	,,	—	180	149		
1914.	Jan.	246	646	,,	72		221		—
	Feb.	228	600	,,	26	—	247		—
	Mar.	193	507	,,	—	67	180		—
	Apr.	150	394	,,	—	180	0 (empty)		
	May	479	1260	,,	686		686		—
	June	452	1190	,,	616	—	1302		—
	July	430	1130	,,	556		1858		—
	Aug.	385	1016	,,	442	—	2300 (full)	—	—
	Sept.	143	376	,,	—	198	2072		
	Oct.	122	321	,,		253	1819	—	
	Nov.	101	266	,,		308	1511		
	Dec.	87	228	,,		346	1165		—


VOLUME c f.s.
500
400
300
200
100
E
F
A
H
B
C
G
E'
F'
J F M A M J J A-S O N D
1911
J F M A M J J A S O N D
1912
J F M A M J J A S O N D
1913
J F M A M-J J A S O N D
1914

area below the line, and the reservoir, if empty at A, will again be empty at C. The line FF′ corresponds to a consumption of 218 c.f.s., in which case the reservoir, if full at B, will again be full at G, and the area above the line between B and G is balanced by the area below the line. Each of these latter areas represents 2300 million cubic feet, giving the required capacity of the reservoir for this state of affairs. Under these conditions the reservoir, if empty at A, will overflow at H, where the area above the line to the left of the vertical through H represents 2300 million cubic feet, and the amount of water wasted will be given by the coarsely hatched area, which represents 1970 million cubic feet.

With no storage, the maximum continuous output would correspond to the minimum ordinate of the hydrograph, which in this case represents a flow of 66 c.f.s. If a larger minimum output is desired than can be obtained with the available storage capacity, this can only be obtained from an auxiliary steam-, gas-, or oil-power plant. Thus if a uniform output equivalent to the full stream capacity of 238 c.f.s. were required, with a storage capacity of 2300 million cubic feet, the capacity of the auxiliary station would be represented by the area shown finely hatched in the diagram. This area occupies 142 squares, representing 1865 million cubic feet. The total output over the four years considered would be 30,000 million cubic feet, so that the necessary output of the auxiliary station would be 6·2 per cent of the total.

If, for example, the working head were 250 ft. and the over-all efficiency of the turbines 80 per cent, the continuous output would be 5400 h.p., and the necessary output of the auxiliary station would average (5400 × ·062 × 24) or 8050 h.p. hours per day. The maximum continuous output with no storage would be 1500 h.p., and with a storage of 2300 million cubic feet and no auxiliary plant would be 4940 h.p.

28. Power Percentage-time Curves.— From the tabulated data or from the graph of fig. 18, a curve can be plotted showing the percentage of total time during which the various flows or corresponding horse-powers are available. In fig. 19 two such curves are plotted for the preceding example, for the cases where the stream-flow is unregulated and where it is regulated by a reservoir of 2300 million cubic feet capacity. The curves show that with the unregulated stream a flow of 350 c.f.s. is available for 20 per cent of the time, 260 c.f.s. for 40 per cent of the time, and so on; while with the regulated stream 218 c.f.s. is available the whole time, and 260 c.f.s. is available for 20 per cent of the time.

29. Auxiliary Plant.—Where storage is impossible or unduly costly, an auxiliary steam-, gas-, or oil-power plant may be installed to work in conjunction with the water-power plant, and to serve in effect as a storage system for the latter. Such an auxiliary installation increases the capacity of the plant, not only by its own output, but because it enables water, which would otherwise be wasted, to be utilized. In fig. 19, for example, a steam installation capable of raising the minimum output to that corresponding to a flow of 150 c.f.s., and itself capable of giving a maximum

output equivalent to a flow of 84 c.f.s., would need to develop an amount of energy represented by the area *efk*. The additional energy developed from water power would be represented by the area *efgh*, and is about 7·5 times as much as that developed by the steam plant.* Where storage is expensive, the combination of a steam- and water-power plant often forms the most economical method of using water power, and the majority of water-power installations in the United States have been developed in this way. The saving due to such an auxiliary installation diminishes as the

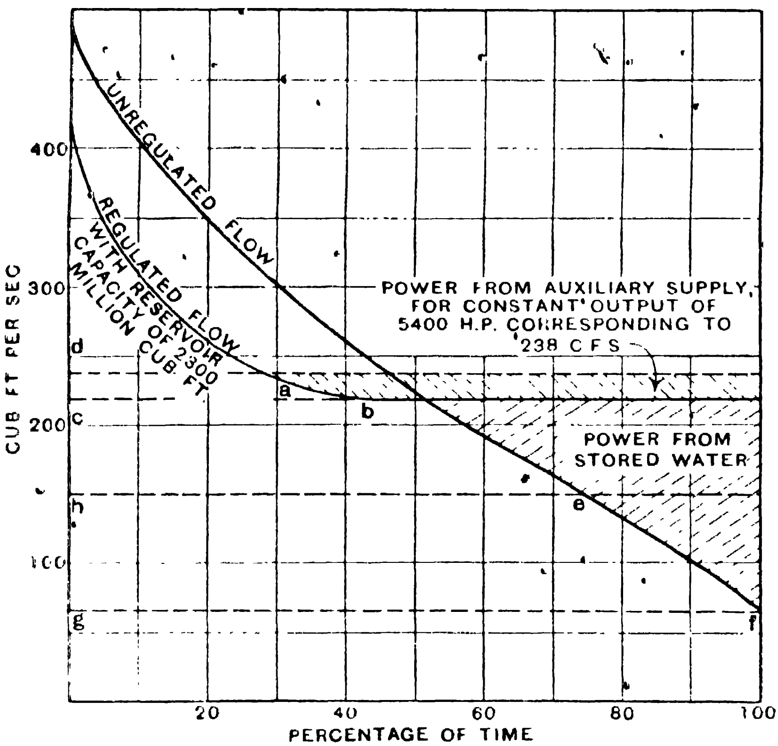


Fig 19.—Power Percentage-time Curve for Stream

storage is increased. Thus with the regulated flow of fig. 19, and with auxiliary power designed to give a constant output corresponding to a flow of 238 c.f.s., the amount of additional energy developed by the water power would only be represented by the area *abcd*.

Fig. 20 shows the amount of energy developed from water power and from the auxiliary steam plant in the case of the British Columbia Electric Railway Co., Ltd., during the years 1908–13. During the period 1909–13 considerable additions were made to the water-power plant.

The most economical size of an auxiliary plant, designed with a view of acting as supplementary to the hydro-electric plant at times of low water

* Department of the Interior, Ottawa. *Water Power Resources*, Paper No. 13, 1915.

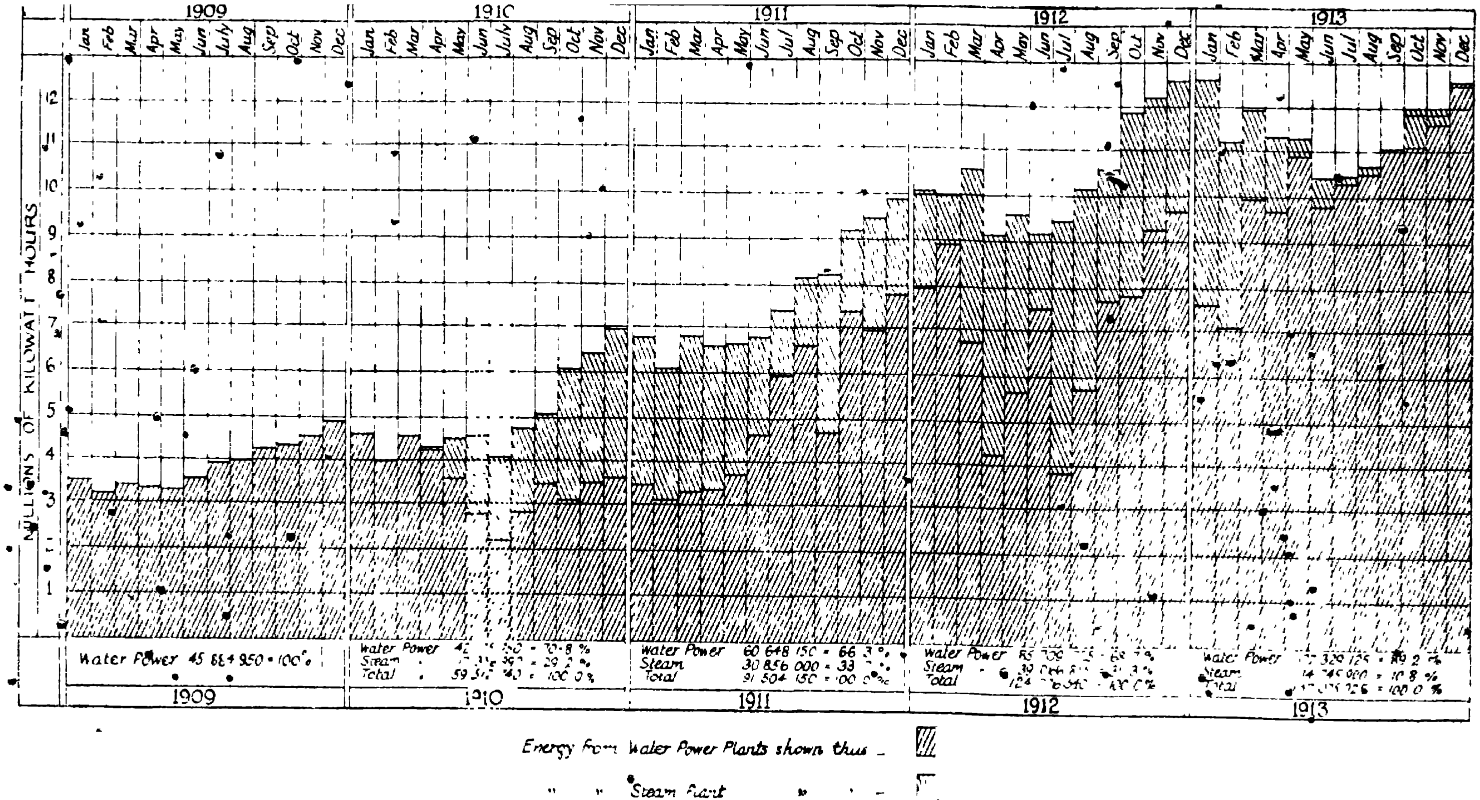


Fig 20.—Energy Developed from Water Power and Steam in Plant of British Columbia Electric Railway Co., Ltd.

or of peak load, can only be estimated by careful investigation of the relative costs of additional storage and of auxiliary power. In some installations in the United States it has been found advisable to install an auxiliary plant having a capacity twice as great as the low-water flow capacity of the water-power plant, but the economical ratio is usually much less than this. The size depends materially on the method of utilization. If the storage be exhausted before the auxiliary plant is started, a plant of large capacity is required, but the period of operation is short. If, on the other hand, the plant be operated whenever the storage becomes low, a smaller capacity is required, but a longer period of operation is necessary. Under such conditions full advantage is seldom taken of the storage capacity, and in consequence a larger number of horse-power hours require to be developed by the auxiliary plant.

Where the minimum stream-flow together with the output of the auxiliary plant is insufficient to carry the peak load, the latter method of operation must be used, the auxiliary plant being operated continuously at such times, and the water power being used to carry the peaks of the load. In general, since for maximum economy a steam plant requires to operate at more or less constant load, corresponding to full load on one or more units, this method of operation is advisable. Where the mean power demand exceeds the output of the hydraulic installation, the auxiliary plant should be operated continuously, the water power being devoted mainly to carrying the peak loads.

Reserve Plant.—In the case of a hydro-electric installation which supplies light and power for general distribution in a city, a reserve steam or gas plant is often installed, and is kept ready for immediate use at any time in case of an accident to the hydro-electric plant or transmission line, or of a serious reduction in its output due to drought, floods, or ice troubles. Where the output is utilized for factory supply or for purely industrial purposes, such a fuel-operated reserve plant is usually considered unnecessary.

Since a reserve plant is only operated at long intervals, the predominating factor in deciding on the type of plant should be low capital costs rather than low operating costs. The plant should be designed with a view of rapid attainment of load on emergency. For large units the steam turbine with water-tube boilers, and for small units the gas- or oil-engine, most nearly satisfy the required conditions.

The capacity of such a plant is governed by the magnitude of the load which must be maintained under all conditions. Whether its services are required or not, it must be run periodically, and the charges to be debited to it add very appreciably to the cost per unit of the power sold. In the case of a steam-turbine plant of, say, 10,000 kw. capacity, the total annual charges without operation under present-day conditions would amount to approximately £35,000.

30. Load Curves.—In any power plant the load varies from hour to hour and from day to day. The "load factor" is the ratio of the average

to the maximum load carried. The load factor and the characteristics of the load curve depend largely on the class of consumer supplied. Thus fig. 21 shows typical curves, AA for a municipal lighting and power supply, and BB for a large industrial factory load. Fig. 22 shows typical winter and summer load curves on the Niagara system.* Here the individual loads are typical commercial, domestic, and municipal loads, and do not include any electro-chemical or -metallurgical processes. Pumping to reservoirs is carried out during off-peak hours, and accounts for the magnitude of the night load. Fig. 23 shows the seasonal variation of the peak

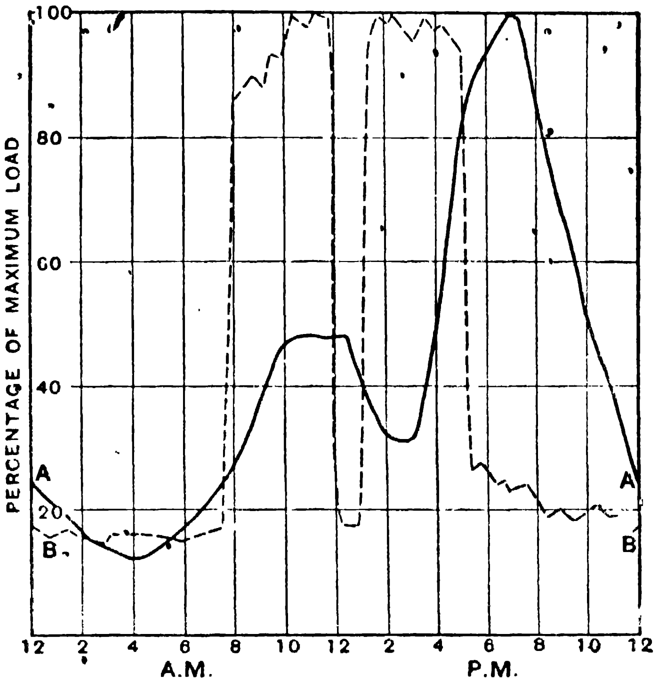


Fig. 21.—Typical Load Curves for Municipal Lighting and Power (AA), and for Industrial Factory (BB)

load on the Central Ontario system,† and also indicates the rate of growth during the period in question.

The design of a hydro-electric installation depends largely on the type of load to be anticipated. Where the load factor is high, as in the case of a plant supplying an electro-chemical or electro-metallurgical industry, or any industry with a large night load, the turbine capacity for a given annual output is much smaller than when the load factor is low; the capital costs are correspondingly lower; and since the capital charges form the largest part of the total cost of the power, the cost per unit is correspondingly lower.

Where a plant is supplied directly from a perennial stream with no

* Department of the Interior, Canada: *Water Resources*, Paper No. 17, p. 47.

† *11th Annual Report of Hydro-Electric Power Commission*, Ontario, 1918, Part I.

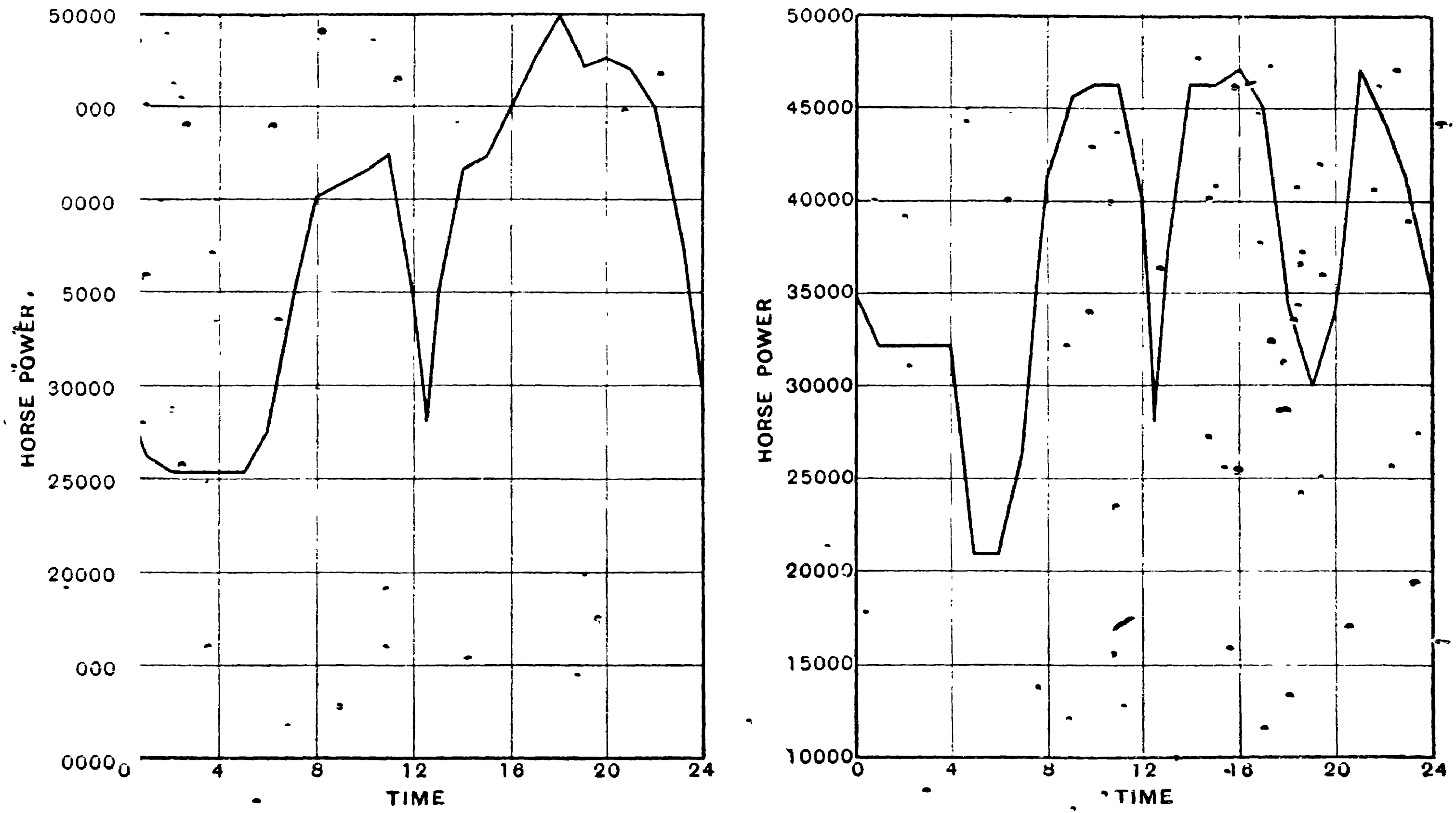


Fig. 22.—Typical Winter and Summer Daily Load Curves on Niagara System

storage, the output from a turbine installation operating at full capacity on the peak load is directly proportional to the load factor, and the charge per unit will require to be inversely proportional to the load factor or very nearly so. In such a case any demand occurring during off-peak hours is a clear gain to the installation in that the only expenses to be debited to this load are those due to the extra attendance and supplies required. Such a load can be economically supplied at a very low rate, and in the interests of the installation every encouragement should be given to such consumers, with a view of making the load factor as nearly as possible unity.

Where the head is high and the plant is designed with sufficient storage

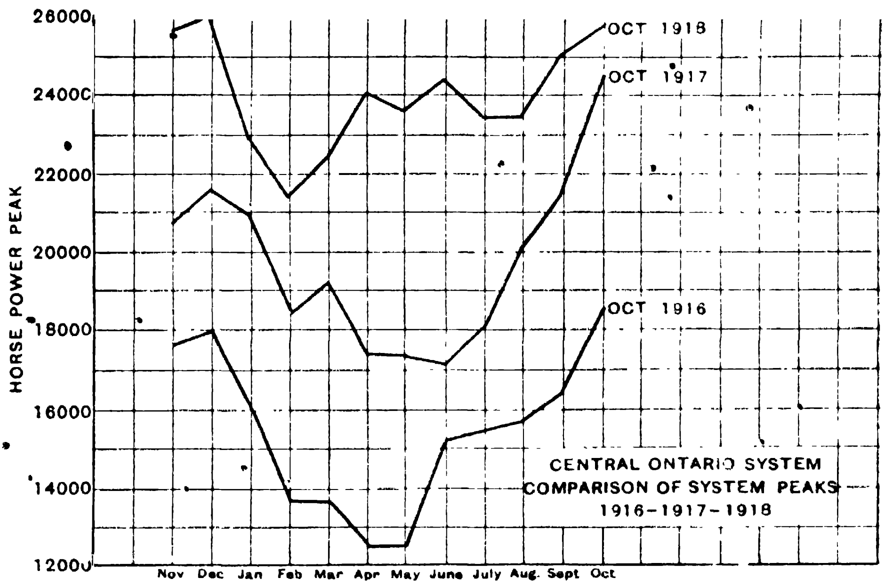


Fig. 3

capacity to utilize the whole or a large proportion of the run-off from a catchment area, the cost of the necessary storage area and dam will be largely independent of the load factor. The head works and canal, flume, or pipe line must, however, be designed to deal with the maximum rate of flow, unless the physical configuration of the district renders it possible to provide sufficient pondage near the power house to take care of the hourly fluctuations of demand. In any case, while the capital costs are increased by a low-load factor, the increase is relatively smaller than when no storage is provided. In fact, when the storage charges form a large percentage of the total costs, it may be more profitable deliberately to lay out the scheme for an ordinary industrial load in spite of its comparatively low-load factor, and to discourage the continuous load, which, from the nature of the case, is seldom capable of bearing the same charges per unit as is the industrial load.

Where the load is very variable, it is necessary for efficient operation that two or more turbines be installed, of such capacity that one unit will carry the minimum load with reasonable efficiency. As the demand increases, successive units are started up, until at maximum load all the units, except those provided as a stand-by, are in operation.

CHAPTER V

Hydraulics

Hydraulics of pipe flow; flow in open channels; weir flow; impounding flood discharge.

31. Water in motion possesses energy in virtue of its velocity, its pressure, and its height. It has kinetic energy, pressure energy, and potential energy. Thus water in motion with velocity v f.s. at the point A (fig. 24) has kinetic energy $v^2 \div 2g$ ft. lb. per pound. Its pressure energy is $p \div w$ ft. lb. per pound where p is its pressure in pounds per square foot, and w its weight per cubic foot, and its potential energy is z ft. lb. per pound where z is its height in feet above some datum level. Each of these expressions is equivalent to a height or head in feet. Thus $v^2 \div 2g$ is the height through which a body falling freely would attain a velocity v, while $p \div w$ is the height of a column of water which would produce the pressure p at its base. $p \div w$ is therefore called the pressure head.

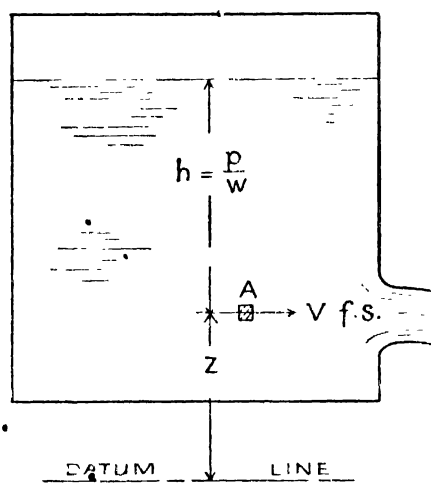


Fig. 24

32. Bernoulli's Theorem.—If water flows from a point (1) to a point (2), and if there is no loss of energy between these points, the relationship

$$\frac{p_1}{w} + \frac{v_1^2}{2g} + z_1 = \frac{p_2}{w} + \frac{v_2^2}{2g} + z_2 = \text{constant}$$

holds. This is known as Bernoulli's Theorem. If, due to wall friction or eddy formation, there is a loss of energy of h_f ft. lb. per pound, which is equivalent to a loss of head of h_f feet between (1) and (2), the equation becomes

$$\frac{p_1}{w} + \frac{v_1^2}{2g} + z_1 = \frac{p_2}{w} + \frac{v_2^2}{2g} + z_2 + h_f.$$

In hydraulic problems, atmospheric pressure is taken as the datum pressure, so that, for example, the water in a parallel jet discharging under atmospheric pressure is taken as having no pressure energy.

33. Hydraulic Losses.—Water flowing over any solid surface, under practical conditions of operation, suffers a loss of energy due to so-called surface friction, which is really due to eddy formation at the surface, and which increases with the roughness of the surface. Also, any change in the direction of flow, or any reduction in velocity, such as occurs when the cross-sectional area of a water passage is increased, is productive of eddy formation and of loss of energy.

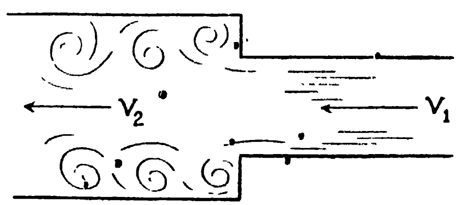


Fig. 25

Loss due to Enlargement of Section.—When a pipe line has its cross-sectional area suddenly increased from A_1 to A_2 sq. ft. (fig. 25), so that the velocity is reduced from v_1 to v_2 f.s., the loss of head is given very closely by the expression

$$\frac{(v_1 - v_2)^2}{2g} \text{ ft.}$$

This loss may be reduced within limits, by reducing the velocity gradually (fig. 26). In this case the loss of head is $k(v_1 - v_2)^2 : 2g$, where k has the following mean values:*

	θ	2°	5	10	15°	20°	30°	40	50°	60°	70°	80	90	120°	150°	180°
Circular Pipe	K	20	13	18	27	43	75	91	1 05	1 12	1 12	1 10	1 07	1 05	1 03	1 00
Rectangular Pipe, with one pair of sides parallel	K		31	18	[illegible]	48	90	1 10	—	—					—	—

These losses include the skin friction in the pipe. This accounts for the value of F increasing as θ is diminished below a definite value, about 6° in a circular pipe, and 11° in a rectangular passage, owing to the increasing length of pipe between points (1) and (2).

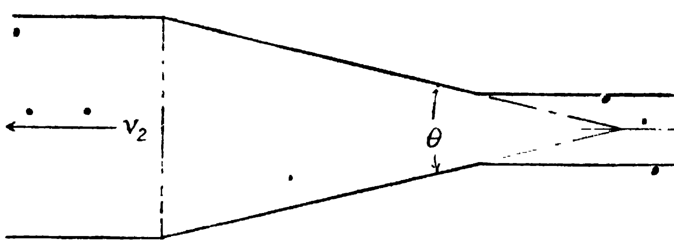


Fig. 26

Loss at Valves and Sluices.—The loss of head due to a partially opened valve is largely due to the expansion of the stream section on passing the constriction. The loss is also affected appreciably by irregularities in

* Gibson, *Hydraulics* (Constable & Co., London), p. 85.

design, so that values deduced from tests on any one design of valve can only be taken as applying approximately to another valve. The following values have been determined experimentally from valves * of the types

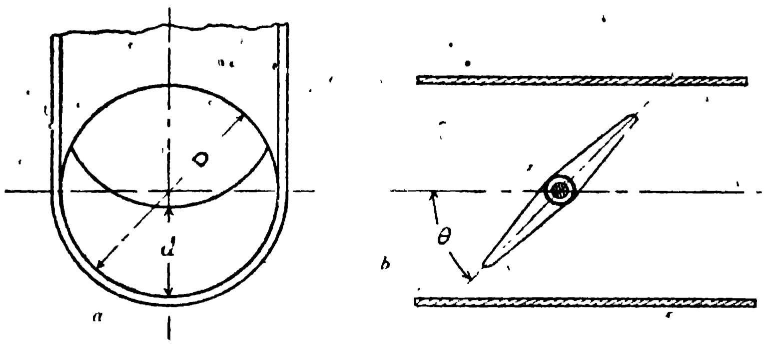


Fig. 27

shown in figs. (27*a* and *b*). Here the loss equals $Fv^2 \div 2g$ ft., where v is the velocity in the pipe

Type of Valve.	$d \div D$									
	·2	·3	·4	·5	·6	·7	·8	·9	1·0	
Circular Sluice Gate (fig. 22*a*), 2″ diam.	30	11	4·2	2·1	·9	·35	·22	·07	·00	Values of F.
Circular Sluice Gate (fig. 22*a*), 24″ diam	36	11	3·0	1·6	1·0	--	--	–	—	Values of F.

	θ								
	5°	10°	20°	30°	40°	50°	60°	70°	
Butterfly Valve (fig. 22*b*)	·24	·52	1·54	3·9	10·8	32·6	118	751	Value of F.

Losses at Bends in a Pipe Line.—The loss due to a right-angled bend depends on the radius of curvature R of the bend.† The best radius in practice is from 2·5 to 5·0 pipe diameters. For such bends the loss is given sufficiently nearly by $\cdot 3v^2 \div 2g$ ft. Where the bend is carried round an angle δ less than 90°, the loss is very nearly proportional to δ^2.

34. Flow in Pipe Lines.—In designing a pipe line, the problem which usually presents itself to the engineer is that of determining the minimum size of pipe which, with a given loss of head, will discharge a given volume of water per second. The available head is absorbed in giving the kinetic energy of flow in the pipe ($v^2 \div 2g$), and overcoming the pipe line losses which are due:

* Gibson, *Hydraulics* (Constable & Co., London), p. 249. † *Ibid.*, p. 251.

1. To eddy formation at the entrance to the pipe;
2. To bends, valves, changes of sections, &c.;
3. To wall friction in the pipe.

The loss due to eddy formation at the entrance is small. With a bell mouthpiece it is about $\cdot05\ v^2 \div 2g$ ft. With a pipe opening flush with the side of the reservoir it is about $\cdot47v^2 \div 2g$ ft., and, with a pipe projecting into the reservoir, about $v^2 \div 2g$ ft.

Friction Losses in Pipe Lines.—Many experiments have been carried out to determine the loss due to wall friction in a straight pipe. The earlier experimenters assumed this to be proportional to v^2, and inversely proportional to the hydraulic mean depth m, which is equal to the cross-sectional area $\div$ wetted perimeter.

On this assumption, the loss in friction is written as

$$h = \frac{flv^2}{2gm}\text{ ft.},$$

or, in the form adopted by Chezy,

$$v = c\sqrt{mi} = c\sqrt{m\frac{h}{l}},$$

where c and f are coefficients whose values depend on the roughness of the pipe. More recent investigations have shown that the coefficient also depends on the pipe diameter and on the velocity of flow. Probably the best known of these latter formulæ of the Chezy type are due to Kutter and Ganguillet, and to Bazin.

Kutter wrote:

$$c = \frac{41\cdot6 + \frac{1\cdot811}{n} + \frac{\cdot00281}{i}}{1 + \left(41\cdot6 + \frac{\cdot00281}{i}\right)\frac{n}{\sqrt{m}}},$$

where $n = \cdot011$ for asphalted cast-iron pipes, very smooth new cast-iron pipes, and cement-lined tunnels;
,, $n = \cdot0125$ for ordinary new cast-iron and wooden pipes;
,, $n = \cdot013$ for pipes of class (1) when incrusted and slightly tuberculated;
,, $n = \cdot014$ for new riveted steel pipes;
,, $n = \cdot015$ for well-laid pipes having a slimy coating and deposits over the surface, or with slight tuberculations;
,, $n = \cdot017$ for heavily tuberculated pipes.

Various tables are available giving values of Kutter's c corresponding to different velocities, pipe diameters, and slopes for the various values of n.

Bazin, from experiments on channel flow, deduced the formula:

$$= \frac{157\cdot6}{\frac{\gamma}{\sqrt{m}}},$$

where γ depends on the roughness of the surface. Mallet,* from an examination of experiments on pipes and pressure tunnels, finds that by writing

$$c = \frac{a}{\beta + \gamma \frac{n}{\sqrt{m}}},$$

the following values of a, β, γ, and n give good results for the following classes of pipe:

Class (1) above: $n = \cdot 011$; $a = 172$; $\beta = 1$; $\gamma = 30$.
„ (3) „ $n = \cdot 013$; $a = 162$; $\beta = 1$; $\gamma = 30$.
„ (5) „ $n = \cdot 015$; $a = 130$; $\beta = \cdot 9$; $\gamma = 26$.
„ (6) „ $n = \cdot 017$; $a = 112$; $\beta = \cdot 6$; $\gamma = 52$.

The most recent investigations tend to show that an exponential formula, of the type

$$h = \frac{flv^n}{d^x} \text{ ft.}$$

more nearly agrees with experimental results. Values of f, n, and x have been determined by many observers.† Mr. A. A. Barnes,‡ in a recent discussion of experiments by himself and other observers, gives the following values as applying to new and cleaned pipes:

Material.	Formulæ for		
	Mean vel., f.s. v	Friction head, h ft. h	Q, c.f.s. Q
New uncoated cast-iron pipes	$136{\cdot}6m^{\cdot 60}i^{\cdot 51}$	$\cdot 000343\frac{lv^{1\cdot 953}}{d^{1\cdot 172}}$	$46{\cdot}7d^{2\cdot 60} \times i^{\cdot 512}$
New asphalted cast iron	$174{\cdot}1m^{\cdot 700}i^{\cdot 529}$	$\cdot 000436\frac{lv^{1\cdot 801}}{d^{1\cdot 454}}$	$47{\cdot}1d^{2\cdot 700} \times i^{\cdot 529}$
New asphalted single-riveted wrought-iron and steel pipes	$171{\cdot}4m^{\cdot 723}i^{\cdot 527}$	$\cdot 000386\frac{lv^{1\cdot 808}}{d^{1\cdot 372}}$	$49{\cdot}4d^{2\cdot 723} \times i^{\cdot 527}$
Do., double-riveted with taper or cylinder joints	$129{\cdot}9m^{\cdot 44}i^{\cdot 52}$	$\cdot 000270\frac{lv^{1\cdot 923}}{d^{\cdot 846}}$	$55{\cdot}4d^{2\cdot 44} \times i^{\cdot 52}$
New smooth wood-stave pipes	$223{\cdot}3m^{\cdot 66}i^{\cdot 586}$	$\cdot 00047\frac{lv^{1\cdot 707}}{d^{1\cdot 126}}$	$71{\cdot}3d^{2\cdot 66} \times i^{\cdot 586}$
New unplaned wood-stave pipes	$182{\cdot}5m^{\cdot 666}i^{\cdot 560}$	$\cdot 000541\frac{lv^{1\cdot 757}}{d^{1\cdot 171}}$	$58{\cdot}5d^{2\cdot 666} \times i^{\cdot 569}$
Clean neat cement pipes	$136{\cdot}3m^{\cdot 635}i^{\cdot 484}$	$\cdot 00024\frac{lv^{2\cdot 066}}{d^{1\cdot 312}}$	$42{\cdot}0d^{2\cdot 635} \times i^{\cdot 484}$

* *Proc. Inst. C. E.*, 1918–9. † Gibson, *Hydraulics* (Constable & Co., 1912), p. 201.
‡ A. A. Barnes, *Hydraulic Flow Reviewed* (Spon, London), 1916.

Owing to the very convenient form of Chezy's equation,

$$v = c\sqrt{mi},$$

it is often an advantage to have at hand values of c corresponding to various diameters and velocities of flow. Such approximate values are given in the following table:

Material.	Velocity, f.s.	Diam. Inches. 6	12	18	24	36	48	60	72	120
New cast iron ..	2	100	107	111	115	120	124	-	—	--
	4	104	111	115	119	124	128	-	—	–
	6	106	113	117	121	126	130	-	—	-
	8	107	114	118	122	127	131		—	-
Clean asphalted pipes; smoothly finished concrete pipes and cement-lined tunnels	2	-	103	108	113	120	126	131	135	139
	4	-	108	113	118	126	132	137	141	145
	6		112	117	122	131	137	142	145	149
	8		115	120	125	134	141	146	149	153
	10		117	122	127	136	143	148	151	155
New single-riveted steel or wrought-iron pipes.. ..	2	-	97	103	108	114	119	123	126	—
	4		103	109	114	120	125	129	132	—
	6	-	107	113	118	125	129	134	137	—
	8		109	115	121	128	133	138	141	—
	10		111	117	123	129	135	140	143	—

Double-riveted steel pipes with cylinder joints have values of c about 5 per cent lower than single-riveted pipes. A new wood-stave pipe has values about 5 per cent lower than a clean asphalted pipe.

After a period of use the incrustation of a pipe line diminishes its discharge. The rate and type of incrustation depends on the class of water and on the material of the pipe walls.

Barnes * found that the percentage diminution in the discharge of an asphalted cast-iron pipe, with diameters from 40 to 44 in., conveying raw moorland water, was given by the relationship:

$$\text{Percentage diminution in discharge} = 13\ (\text{age of pipe in years})^{0.37}.$$

Thus after one year the discharge had diminished by 13 per cent, although probably no nodules of incrustation had been formed at this stage. The corresponding diminution after ten years was 30·5 per cent, and, after 20 years, 39·4 per cent. The rate of diminution is much less in a wooden pipe, or in a concrete or cement-lined pipe or tunnel, than in an iron or steel pipe. To allow for this diminution, the pipe should be designed to give an initial discharge in excess of the requirements.

* *Proc. Inst. C. E.* 1918-9.

The excess percentage discharge for different types of pipe should be approximately as follows:*

Type of Pipe.	Uncoated Cast Iron.	Asphalted Cast Iron.	Asphalted Riveted Wrought-iron or Steel Pipes.	Wood Stave.	Neat Cement or Concrete.
Discharge for which designed, in terms of desired discharge Q.	1·55 Q	1·45 Q	1·33 Q	1·08 Q	1·66 Q

No data is available as to the relative diminution in the case of a large and a small pipe. It must evidently be greater in a small pipe, and the values in the preceding table are therefore to be applied with caution. They are probably valid for pipes over 24 in. in diameter, which may be required to operate for ten years before being cleaned out.

35. Working Velocities of Flow.—By adopting a high velocity of flow, the diameter and cost of the pipe line is reduced, but the friction loss and the difficulty of speed regulation are increased. In a plant supplied through a long closed penstock under a high head, considerations of speed regulation will usually be most important, while where the head is moderate and a stand pipe can be installed, the friction loss will be the more important factor. The most economical pipe line is one in which the sum of its annual charges due to interest, maintenance, and depreciation, and of the value of the power lost per annum in friction, is a minimum.

The permissible velocity depends largely on the load factor of the plant. The pipe feeding a turbine which operates normally at part load, or only occasionally requires to develop full load, can be designed to give a relatively high velocity at full load without the friction loss under normal operation becoming excessive. Generally speaking, a lower velocity should be adopted for a low-head development than for a high-head development. In practice, velocities of from 6 to 12 ft. per second are usual.

36. Hydraulic Gradient.—If, as in fig. 28, a horizontal AB be drawn through the free water surface, and if ordinates be drawn downwards from AB to represent, on the vertical scale of the drawing, the total loss of pressure head from the pipe entrance to the particular point considered, the ends of such ordinates, being joined, give a curve called the *hydraulic gradient* for the pipe. If a series of open stand pipes were erected on the pipe line, the free surfaces in these pipes would lie on the gradient line, and the pressure in the pipe is represented, at each point, by its distance below this line. If the pipe is above the gradient line at any point, the pressure will be less than atmospheric. In order to prevent difficulties arising from liberation and accumulation of air at such points, and from admission of air at leaky joints, the greatest height above the gradient line should never exceed 20 ft. Owing to the weakness of large pipes under

* Barnes, *Hydraulic Flow Reviewed.*

external pressures, such pipes should not be laid above the gradient line.

37. Flow through Pipes coupled up in Parallel.—If a series of

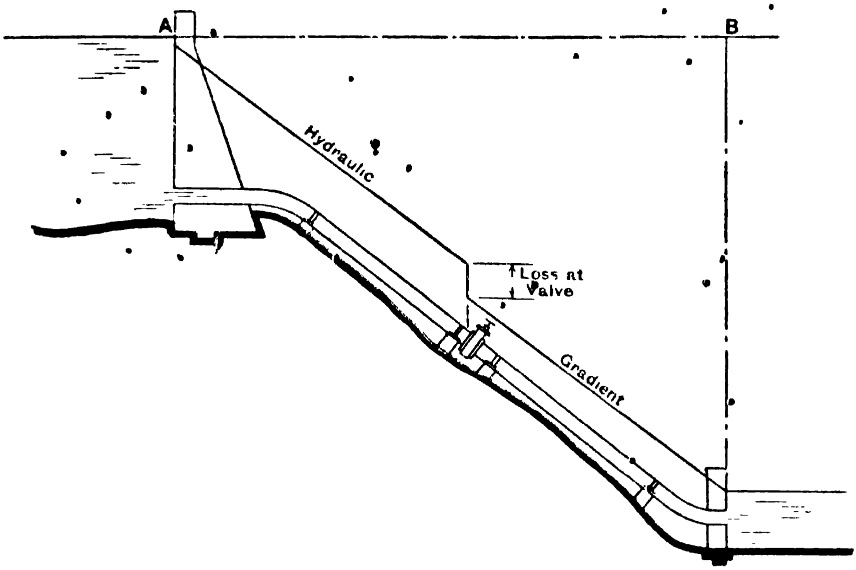


Fig. 28.—Hydraulic Gradient of Supply-pipe Line

pipes of diameters d_1, d_2, &c., discharge in parallel between the same two points, so that the available head h is the same in each case, adopting the relationship

$$h = \frac{klv^n}{d^x},$$

the total flow Q, which equals $\frac{\pi}{4}\{v_1d_1^2 + v_2d_2^2 + \text{\&c.}\}$ c.f.s., becomes

$$Q = \frac{\pi}{4}\left(\frac{h}{k}\right)^{\frac{1}{n}}\left\{\frac{d_1^{2+\frac{x}{n}}}{l_1^{\frac{1}{n}}} + \frac{d_2^{2+\frac{x}{n}}}{l_2^{\frac{1}{n}}} + \text{\&c.}\right\}$$

$$= \frac{\pi}{4}\left(\frac{h}{k}\right)^{\frac{1}{n}}\Sigma\left(\frac{d^{2+\frac{x}{n}}}{l^{\frac{1}{n}}}\right) \text{ c.f.s.}$$

E.g. taking, as for a cast-iron pipe, $n = 1{\cdot}953$, $x = 1{\cdot}172$,

$$Q = \frac{\pi}{4}\left(\frac{h}{k}\right)^{\cdot 512}\Sigma\left(\frac{d^{2\cdot 60}}{l^{\cdot 512}}\right).$$

Thus three small pipes of diameter d will give the same discharge as a single large pipe D, of the same length, if

$$3d^{2\cdot 60} = D^{2\cdot 60},$$

i.e. if $$D = 3^{\frac{1}{2\cdot 6}}d = 1{\cdot}53d$$

or one pipe 36·7 in. in diameter would give the same discharge as three 24-in. pipes.

Since, for large pipes, under fairly high pressures, the cost of the pipe line laid complete is very approximately proportional to $d^{1\cdot 5}$, the cost of the three small pipes would be 60 per cent. greater than that of the large pipe. From this it is clear that, in general, the number of pipes in parallel should be the minimum compatible with satisfying conditions of operation and of supply. This statement is to be qualified in the case of a long supply line under a low head. Here the pipe walls, if designed simply to withstand the internal pressure, would, if the pipes were very large, be inadequate to resist the distorting forces due to their own weight and that of the contained water. The necessity for increasing their wall thickness adds to their cost, and in such a case the most economical subdivision of the pipe line must be decided from a consideration of the special circumstances.

38. Long Pipe Line, terminating in a Nozzle.—Let A be the area, D the diameter, and V the velocity of flow in the pipe line, and let a, d, and v refer to the nozzle. Thus if h be the available head, and if the Chezy formula be adopted, we have, in a long pipe line:

$$h = \frac{4V^2 l}{C^2 D} + \frac{v^2}{2g}$$

$$= \frac{v^2}{2g}\left\{\frac{8gld^4}{C^2D^5} + 1\right\}, \text{ since VA} = va.$$

$$\therefore v = \sqrt{\frac{2gh}{1 + \frac{8gl}{C^2}\cdot\frac{d^4}{D^5}}} \text{ ft. per second.}$$

In general, the coefficient of velocity, Cv, of a well-designed Pelton wheel nozzle is about ·985, and the velocity will be reduced in this ratio.

Since the energy discharged at the nozzle per second

$$= \frac{wav^3}{2g} \text{ ft. lb.,}$$

the horse-power delivered at the nozzle is

$$\frac{wa}{2g \times 550}\left\{\frac{Cv\sqrt{2gh}}{1 + \frac{8gl}{C^2}\cdot\frac{d^4}{D^5}}\right\}^3.$$

It may be shown that this expression is a maximum when

$$\frac{d}{D} = \sqrt[4]{\frac{DC^2}{16gl}},\text{*}$$

all dimensions being in feet.

With a jet diameter greater than is given by this ratio, any further opening of the nozzle will result in less energy being given to the wheel,

* Gibson, *Hydraulics*, 1912, p. 284.

which then becomes impossible to govern on part loads. While this can only occur in comparatively long small pipe lines, the point should not be overlooked in the preliminary design.

39. Accelerated and Retarded Flow in Pipe Lines: Water Hammer.—Where, owing to the gradual stoppage of flow at the lower end of a pipe line, the velocity of the water column is gradually reduced, the retardation being a ft. per second per second, this is accompanied by a rise in pressure at the valve of magnitude $\frac{wla}{g}$ lb. per square foot, or of $\frac{la}{g}$ ft. of water.

If the valve closure is sudden, the elasticity of the water is involved. Each layer in turn is brought to rest, its kinetic energy is converted into strain energy, and the disturbance is propagated back to the open end of the pipe with the velocity of sound waves through the medium. Under these conditions, the phenomenon is known as *water hammer*, and the rise in pressure p at the valve is obtained from the relationship

$$\frac{v^2}{2g} = \frac{p^2}{2Kw},$$

$$\text{or } p = v\sqrt{\frac{Kw}{g}} \text{ lb. square foot.}$$

Here K is the modulus of compressibility of the water, which has a mean value of $43 \cdot 2 \times 10^6$ lb. per square foot. Adopting this value, $p = 63 \cdot 7v$ lb. per square inch, a value which shows that excessively high pressures may be obtained with comparatively low velocities of flow, where this action is set up. In a non-rigid pipe line energy is expended in stretching the pipe walls, and the hammer pressure is reduced. Taking this into account, K′, the effective value of K, is given by

$$\frac{1}{K'} = \frac{1}{K} + \frac{r}{2tE}\left(5 - \frac{4}{\sigma}\right),$$

where r is the radius and t the thickness of the pipe, and, for steel pipes, $E = 43 \cdot 2 \times 10^8$ lb. per square foot and $\sigma = 3 \cdot 6$.

It may be shown that pressures as great as those corresponding to instantaneous closure are attained if the time of valve closure does not exceed $2l \div V_p$ sec., where V_p, the velocity of propagation of sound waves along the pipe line, is given by $V_p = \sqrt{\frac{Kg}{w}}$, and is approximately 4700 ft. per second for a rigid pipe line, and may be as low as 3000 ft. per second for a large thin-walled pipe line. If the time of closure is greater than $4l \div V_p$ sec., the formula $p' = \frac{wla}{g}$ lb. per square foot is applicable.*

If the valve is being opened, so that the water column is being accelerated, a corresponding drop in pressure occurs at the valve. These

* For a discussion of the rise in pressure with uniform and non-uniform valve closure, see *Hydraulics*, Gibson, pp. 222–43

changes of pressure are superposed on those due to pipe friction at the velocity obtaining in the pipe line at any instant.

40. Flow in Open Channels.—As in the case of pipe flow, the earlier experimenters assumed the loss of head during steady flow in an open channel to be proportional to the square of the velocity, and adopted one or other modification of the Chezy formula

$$v = c\sqrt{mi}$$

where m is the hydraulic mean depth

$$= \text{cross-sectional area (A)} \div \text{wetted perimeter (P)},$$

and i is the gradient of the channel.

The best-known of these formulæ are due to Ganguillet and Kutter, and to Bazin.

Ganguillet and Kutter put

$$c = \frac{41{\cdot}66 + \frac{1{\cdot}811}{n} + \frac{{\cdot}00281}{i}}{1 + \left(41{\cdot}66 + \frac{{\cdot}00281}{i}\right)\frac{n}{\sqrt{m}}}$$

and Bazin put

$$c = \frac{157{\cdot}6}{1 + \frac{\gamma}{\sqrt{m}}}$$

The values of γ and n in these formulæ depend on the roughness of the surface. For straight channels the following values are applicable:

	Character of Surface.	Bazin's γ	Kutter's n
A	Smooth cement or planed timber.	·109	·009–·010
B	Unplaned timber, slightly tuberculated iron, ashlar, and well-laid brickwork	·290	·012–·013
C	Rubble masonry and brickwork in an inferior condition; fine well-rammed gravel	·833	·017
D	Rubble in inferior condition; canals with earthen beds in perfect condition		·020
E	Canals with earthen beds in good condition	1·54	·0225

Bazin's formula appears on the whole to be the more reliable for artificial channels and conduits, and the corresponding values of c in Chezy's formula for the different surfaces and hydraulic mean depths are given in the following table:

Surface.	Values of c computed from Bazin's Formula.				
	$m = \cdot5$.	$m = 1\cdot0$.	$m = 2\cdot0$.	$m = 5\cdot0$.	$m = 10$.
A	137	142	146	150	152
B	112	122	131	140	145
C	72	86	100	115	125
D	61	74	88	104	115
E	50	62	76	93	106

More recent investigations indicate that a more correct formula for channel flow is of the form

$$v = km^x i^y,$$

where the indices x and y, instead of each being ·50 as in the Chezy formula, have other values.

Different investigators have determined the values of k, x, and y for various surfaces. Among these, Unwin, Thrupp, Fidler, Williams, Tulton, Lea, and Barnes are most worthy of note. Barnes,* as a result of an extended investigation of results by many observers, gives the following values, and formulæ, for clean surfaces:

Surface.	k	x	y	Friction Head h in Feet.	Hydraulic Mean Depth m in Feet.
Planed wooden flumes ..	223·3	·660	·586	$\dfrac{\cdot0000098lv^{1\cdot707}}{m^{1\cdot126}}$	$\dfrac{\cdot000276v^{1\cdot515}}{i^{\cdot888}}$
Unplaned wooden flumes	182·5	·666	·569	$\dfrac{\cdot000107lv^{1\cdot757}}{m^{1\cdot171}}$	$\dfrac{\cdot000403v^{1\cdot502}}{i^{\cdot854}}$
Neat cement channels ..	136·3	·635	·484	$\dfrac{\cdot000039lv^{2\cdot066}}{m^{1\cdot312}}$	$\dfrac{\cdot000435v^{1\cdot575}}{i^{\cdot762}}$
Smooth-faced concrete ..	95·1	·567	·471	$\dfrac{\cdot000063lv^{2\cdot123}}{m^{1\cdot204}}$	$\dfrac{\cdot000324v^{1\cdot764}}{i^{\cdot831}}$
Hard, well-pointed brick..	92·1	·602	·466	$\dfrac{\cdot000061lv^{2\cdot146}}{m^{1\cdot292}}$	$\dfrac{\cdot000546v^{1\cdot661}}{i^{\cdot774}}$
Dressed masonry channels set in cement with no projections	109·7	·713	·483	$\dfrac{\cdot000060lv^{2\cdot07}}{m^{1\cdot476}}$	$\dfrac{\cdot00138v^{1\cdot403}}{i^{\cdot677}}$
Hammer - dressed dry masonry	70·0	·820	·500	$\dfrac{\cdot000204lv^{2\cdot00}}{m^{1\cdot64}}$	$\dfrac{\cdot0056v^{1\cdot22}}{i^{\cdot61}}$
Earth canals in average condition free from vegetation	58·4	·694	·496	$\dfrac{\cdot000275lv^{2\cdot016}}{m^{1\cdot399}}$	$\dfrac{\cdot00285v^{1\cdot441}}{i^{\cdot715}}$

* A. A. Barnes, *Hydraulic Flow Reviewed* (Spon & Co., 1916), p. 47.

of flow depends on the tendency to erosion of the sides and bed. The following safe mean velocities are taken from values given by Ganguillet and Kutter, who, however, state that these are too small rather than too large.

Material of Channel.	Safe Mean Velocity in Feet per Second.
Very soft silty soil	·33
Soft loam	·66
Sand	1·32
Gravel	2·64
Gravel and small pebbles ..	3·94
Conglomerate	6·56
Stratified rock	8·20
Hard rock..	13·2

R. G. Kennedy,* from observations on irrigation canals in light soils, concludes that the safe velocity increases with the depth according to the law $V \propto d^{\cdot 64}$. For sandy soil he gives the values

Depth, feet	2	3	4	5	6	7	8
Mean velocity, f.s. ..	1·3	1·7	2·0	2·4	2·65	2·9	3·2

More recent work shows that for medium depths in light soil a mean velocity of from 1·2 to 1·8 f.s. is safe, while in firm loamy soil the safe

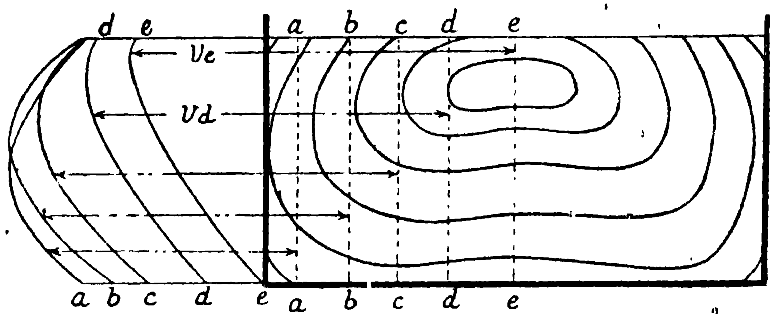


Fig 30. –Contours of Equal Velocity in an Open Channel

velocity is from 3·0 to 3·5 f.s. On firm well-rammed gravel this may be increased to between 5 and 7 f.s. In a concrete-lined channel faced with cement, the maximum safe velocity with water which carries solid material in suspension is about 9 f.s. A higher velocity wears and roughens the bottom until this roughness reduces the velocity sufficiently to prevent further erosion. With a brick or dry-laid heavy rubble channel the

* *Proc. Inst. C. E.*, Vol. 110, 1874–5, p. 281. U. S. Geological Survey: *Water Supply and Irrigation*, Paper No. 95, pp. 76 and 77.

velocity should not exceed 15 f.s. Any higher velocity necessitates a carefully-laid facing of heavy masonry with cemented joints.

44. Distribution of Velocity in an Open Channel.—The distribution of velocity in a straight channel depends somewhat upon the mean

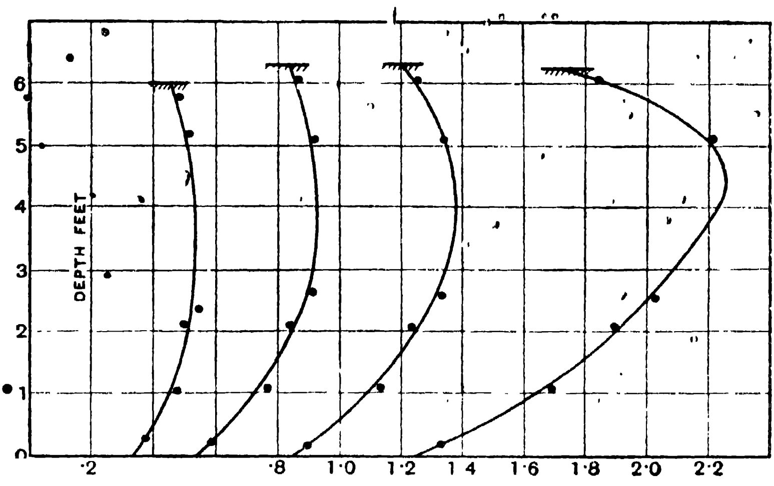


Fig. 31.—Vertical Velocity Curves in Open Channel

velocity. The maximum velocity is found near the centre and in general below the surface, even with a down-stream wind. Its depth usually varies from $\cdot 1h$ to $\cdot 4h$, where h is the depth of the stream. The curves of fig. 30 show typical contours of equal velocity, and the distribution of

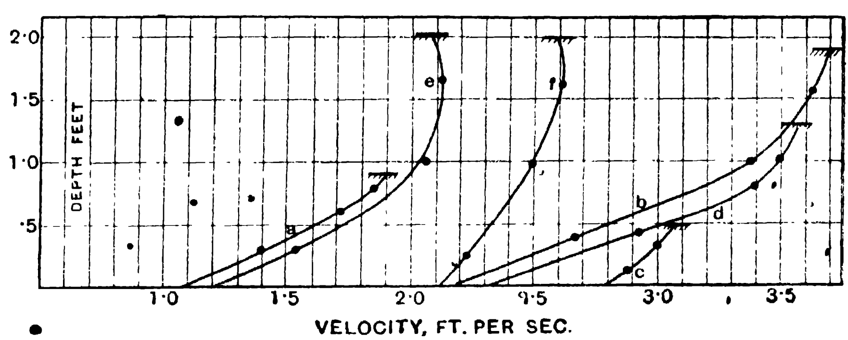


Fig. 32. Vertical Velocity Curves in Open Channel

velocity in a series of verticals in a rectangular channel. Figs. 31 and 32 show the results of a series of gaugings on a concrete channel 16 ft. wide. These curves show the variations of velocity in a vertical plane. Velocity measurements were made at eight equidistant points in a cross section, and each plotted point represents the mean of all eight observations at that particular depth. The effect of an increase in mean velocity in raising

the filament of maximum velocity is well shown by the curves of fig. 31, while the effect, in the same direction, of a diminution in the depth of the stream is apparent from a comparison of the curves of figs. 31 and 32.

It is found that the depth of the point of mean velocity in any vertical is sensibly independent of the direction of the wind, probably because an up-stream wind banks up the head waters and so increases the surface gradient of the stream, thereby increasing the velocity of flow over its lower portions to an extent which compensates for the reduced velocity of the surface layers. The depth of this point varies from about ·55*h*. to ·70*h*, depending on the depth and roughness of the channel as indicated below.

Condition of Bed	Gravel and Small Boulders.				Small Gravel and Sand.				Wood or Cement.			
Depth, feet ..	0 to 2	2 to 4	4 to 6	6 to 10	0 to 2	2 to 4	4 to 6	6 to 10	0 to 2	2 to 4	4 to 6	6 to 10
Depth of point of mean velocity, in terms of *h*.	54	58	·62	66	57	60	65	69	·61	·65	68	70

Generally speaking, the velocity at six-tenths depth in any vertical gives the mean velocity in that vertical within 5 per cent except in abnormal cases, while the mean of the velocities at one-fifth and four-fifths of the depth also gives the mean velocity within narrow limits. The velocity at mid depth is from 1·02 to 1·06 times the mean velocity, the ratio increasing with the depth of the stream. While the surface velocity should only be used for gauging purposes when other measurements are impracticable, its value, on a still day, is between 80 and 100 per cent of the mean velocity in its own vertical. This factor increases with the depth of the stream and with the smoothness of the channel.

In a very shallow stream, the vertical velocity curves approximate to straight lines, and the mean of surface and bottom velocities gives a close approximation to the mean velocity. The ratio of the velocity at mid depth to the mean velocity in any vertical varies very little, its usual value ranging from 1·02 to 1·06, increasing with the depth of the stream. A value of 1·04 will give the mean velocity within 3 per cent for all normal sections and velocities of flow.

The essential results of a number of river gaugings by members of the United States Geological Survey are given below.* Here the symbol S means sandy; G, gravelly; R, rocky; B, boulders; M, mud.

45. Non-uniform Flow in an Open Channel. Under conditions of steady flow in a channel of uniform slope and section, the velocity adjusts itself until the condition $V = c\sqrt{mi}$ is satisfied. Under such conditions the depth assumes a steady value H, such that the discharge $Q = VbH$.

If, owing to some obstruction or irregularity in the channel the depth is altered, the surface slope is also altered. The effect of any such obstruc-

* U. S. Geological Survey *Water Supply and Irrigation*, Paper No. 95, pp. 150–8.

River.	Approximate Width, Feet.	Range of Depth, Feet.	Coefficient for reducing to mean velocity in any vertical the vel. observed at following points.			Per cent of depth at which thread of mean vel. is found.	Character of bottom.
			Six-tenths depth	Mean of top and bottom.	Top.		
Appomattox, Va.	150	2·5— 4·9	1·00	1·11	·81	61·5	S
James, Va.	850	3·0— 4·4	1·01	1·08	·88	59·6	S
Roanoke, Va.	110	1·8— 3·3	·99	1·01	·81	60·9	R and S
Staunton, N.C.	400	2·0— 6·0	·96	1·12	·95	65·7	G and S
Dan, N.C.	600	1·4— 3·7	·99	1·01	·86	61·6	S
Dan, Va.	350	3·5— 9·8	·98	1·09	·90	63·6	S
Reddie, N.C.	85	1·7— 2·5	1·02	·97	·83	57·7	S
Yadkin, N.C.	160	2·7— 6·3	·96	1·12	·91	66·5	S and B
Yadkin, N.C.	450	4·0—11·3	1·02	1·06	·81	59·2	S and R
Catauba, N.C.	200	1·8— 5·0	1·00	1·04	·82	60·6	S and G
Catauba, N.C.	200	6·3— 7·0	1·00	1·08	·88	58·5	M
Catauba, N.C.	110	3·5	1·01	1·05	·80	59·0	S
Wateree, S.C.	300	12·3—17·7	1·01	1·13	·90	58·4	M
Broad, S.C.	500	5·0— 8·9	·96	1·22	·89	65·4	M and S
Saluda, S.C.	800	3·0— 8·5	·97	1·03	·82	62·2	S
Little Tennessee, N.C.	660	3·6— 6·0	1·03	1·06	·83	58·7	B
Nolichucky, Tennessee	300	1·6 — 5·1	1·02	1·02	·80	59·1	B
Fishkill, N.Y.	90	2 — 5	1·036	—	·79	58·7	G
Wallkill, N.Y.	130	3 — 17	·98	—	·85	63·7	S
Farad Flume, Cal. ..	10·1	5·98	·95	1·25	1·09	76·0	Wood
Cornell Canal	16·0	7·2— 8·3	—		—	65·7	Concrete
Cornell Canal	16·0	·55— 1·9	—		—	54·3	Concrete

tion depends on the slope and roughness of the channel, and on the extent to which the depth is initially affected. If, for example, i is less than $g \div c_2$, which, in a channel having $c = 110$, would correspond to a slope less than ·003, the states of affairs produced (a), by a sluice gate reducing the depth to a value less than $H \times \sqrt[3]{\frac{ic^2}{g}}$; ($b$), by a drop in the bed of the stream reducing the depth to between H and $H \times \sqrt[3]{\frac{ic^2}{g}}$; ($c$), by a dam raising the level to a value greater than H, are shown in fig. 33, a, b, and c. If i is greater than $g \div c^2$, the states of affairs produced (d) by a sluice gate reducing the depth to a value less than H; (e) by a drop in the bed reducing the depth to between $H \times \sqrt[3]{\frac{ic^2}{g}}$ and H; and (f) by a dam increasing the depth to above $H \times \sqrt[3]{\frac{ic^2}{g}}$, are shown in fig. 33, d, e, and f.

Wherever the depth attains a critical value, equal to $H \times \sqrt[3]{\frac{ic^2}{g}}$, the surface slope becomes infinite, the profile is perpendicular to the bed of the stream, and a standing wave or waves are produced, as indicated in fig. 33, a, b, e, and f.

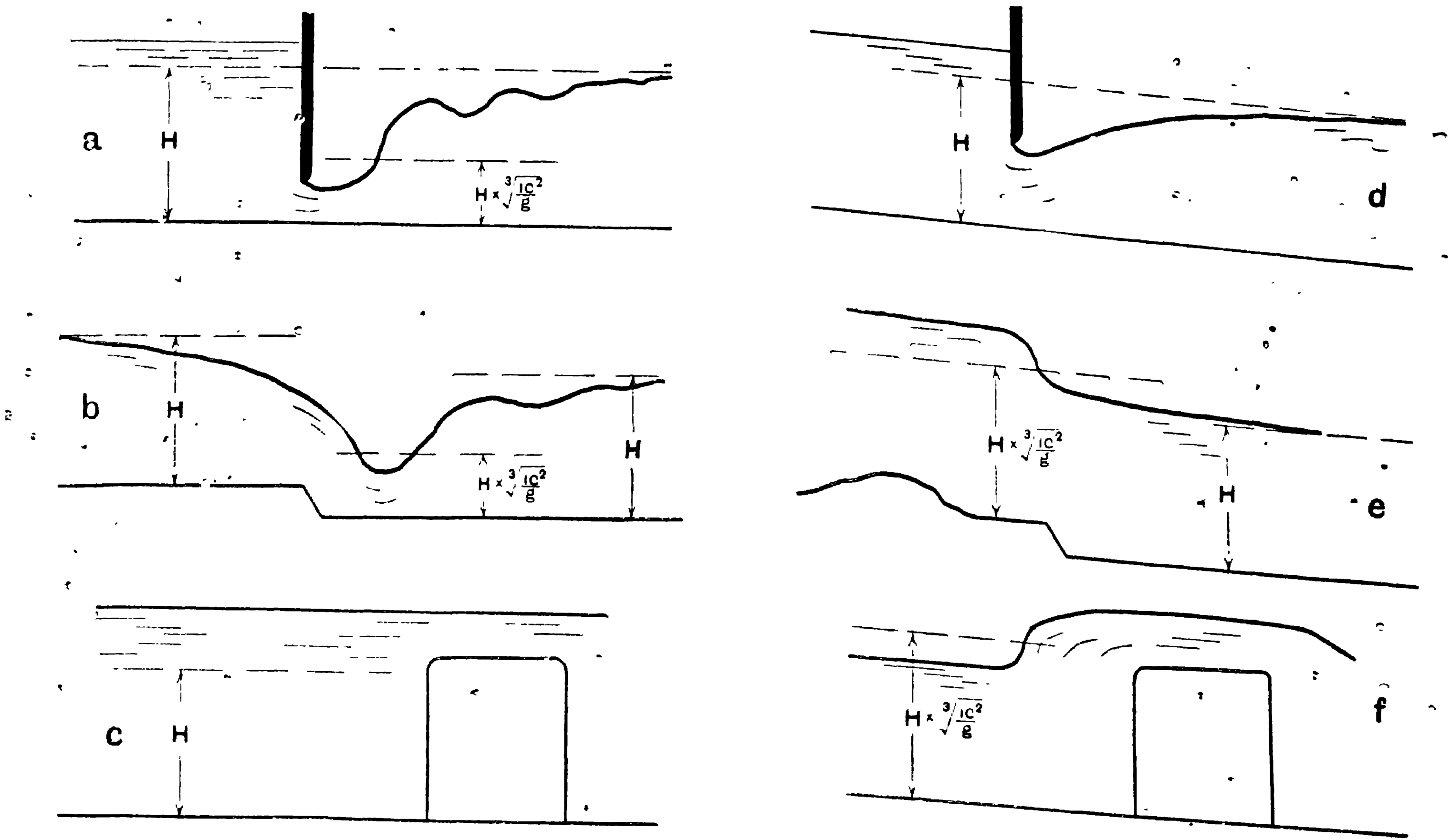


Fig 33 —Stream Profiles in Non-uniform Flow

It will be evident that no gauging operations should be carried out in a portion of a stream likely to be affected by any such disturbances.

Back-water Curve.—The case shown in fig. 33*c*, in which a dam erected across a stream of moderate or small gradient may raise the surface level for a very considerable distance up-stream, is one of importance in practice. If the stream is of regular section, and is wide compared with its depth, it may be shown that the depth h_1, at a point l_1 feet up-stream from the dam, is given by *

$$h_1 - h_0 = il_1 - \mathrm{H}\left\{1 - \frac{c^2 i}{g}\right\}\left\{\phi\left(\frac{h_0}{\mathrm{H}}\right) - \phi\left(\frac{h_1}{\mathrm{H}}\right)\right\}, \quad \ldots\ldots \text{(1)}$$

where h_0 is the depth behind the dam;

where H is the depth of a stream which, if of the same breadth but of uniform depth, would give the same discharge. This makes

$$\mathrm{Q} = \mathrm{A} \cdot c\sqrt{\mathrm{H}i}, \text{ or } \mathrm{H} = \mathrm{Q}^2 \div \mathrm{A}^2c^2i.$$

Values of the function $\phi\left(\frac{h}{\mathrm{H}}\right)$ are given in the following table for various values of the ratio $h \div \mathrm{H}$.

$\frac{h}{\mathrm{H}}$	$\phi\left(\frac{h}{\mathrm{H}}\right)$	$\frac{h}{\mathrm{H}}$	$\phi\left(\frac{h}{\mathrm{H}}\right)$	$\frac{h}{\mathrm{H}}$	$\phi\left(\frac{h}{\mathrm{H}}\right)$	$\frac{h}{\mathrm{H}}$	$\phi\left(\frac{h}{\mathrm{H}}\right)$
1·000	∞	1·020	2·098	1·10	1·587	2·20	1·015
1·001	3·090	1·025	2·025	1·15	1·468	2·50	·989
1·002	2·860	1·030	1·966	1·20	1·387	3·0	·963
1·003	2·725	1·036	1·908	1·30	1·280	4·0	·939
1·004	2·629	1·044	1·843	1·40	1·211	5·0	·927
1·005	2·555	1·050	1·803	1·50	1·162	7·0	·915
1·007	2·445	1·056	1·763	1·60	1·125	10·0	·911
1·010	2·326	1·060	1·745	1·70	1·096	15·0	·909
1·012	2·266	1·070	1·697	1·80	1·073	20·0	·908
1·015	2·192	1·080	1·656	2·00	1·039	50·0	·907

Since this formula is based on the assumptions that c remains constant in spite of the variable depths of the channel, and that the breadth is so relatively large that the hydraulic mean depth is sensibly equal to the actual depth, its results are not strictly applicable to the case of a channel of normal dimensions.

If, however, the value of H be known for the actual channel, the results as calculated from the formula are very nearly in agreement with those obtained by more cumbrous methods in which the variation of c with depth is taken into account, even for extreme sections. As an example, consider the case of a circular concrete conduit 7·30 ft. in diameter, having a gradient of 1 in 3168 or 20 in. per mile, discharging 30 million gallons per day at a normal central depth of 3·40 ft. Let the depth be increased to 5·48 ft. by means of a weir, and determine the depth at a point 4460 ft. up-stream.

* Gibson, *Hydraulics* (Constable & Co., 1912), p. 316.

Before the introduction of the weir

$H = 3{\cdot}40.$

The cross-sectional area $= 19{\cdot}2$ sq. ft.

The wetted perimeter $= 10{\cdot}9$ sq. ft.

The hydraulic mean depth $= 19{\cdot}2 \div 10{\cdot}9 = 1{\cdot}76$ ft.

The mean velocity of flow $= 2{\cdot}90$ f.s.

$$\therefore c = v \div \sqrt{mi} = 2{\cdot}90 \div \sqrt{1{\cdot}76 \times {\cdot}0003157} = 123{\cdot}5.$$

Then from formula (1) p. 67,

$$h_1 - 5{\cdot}48 = ({\cdot}0003157 \times 4460) - 3{\cdot}40\left\{1 - \frac{15250}{3168 \times 32{\cdot}2}\right\}$$

$$\left\{\phi\left(\frac{5{\cdot}48}{3{\cdot}40}\right) - \phi\left(\frac{h_1}{3{\cdot}40}\right)\right\}.$$

$$\therefore h_1 - 5{\cdot}48 = 1{\cdot}410 - 2{\cdot}89\left\{\phi(1{\cdot}612) - \phi\left(\frac{h_1}{3{\cdot}40}\right)\right\}.$$

$$\therefore h_1 - 5{\cdot}48 = 1{\cdot}410 - 2{\cdot}89\left\{1{\cdot}121 - \phi\left(\frac{h_1}{3{\cdot}40}\right)\right\},$$

$$\text{or } h_1 - 2{\cdot}89\,\phi\left(\frac{h_1}{3{\cdot}40}\right) - {\cdot}83 = 0.$$

This equation is to be solved by a process of trial and error.

Thus assuming $h_1 = 4{\cdot}0$; $h_1 \div 3{\cdot}40 = 1{\cdot}178$, $\phi(h_1 \div 3{\cdot}40) = 1{\cdot}425$, and the left-hand side of the equation equals $- {\cdot}94$.

Assuming $h_1 = 5{\cdot}0$; $h_1 \div 3{\cdot}40 = 1{\cdot}47$, $\phi(h_1 \div 3{\cdot}40) = 1{\cdot}177$, and the left-hand side of the equation equals $+ {\cdot}77$.

Evidently the true solution is somewhere between 4·0 and 5·0, and would appear to be slightly nearer 5·0 than 4·0.

Assuming $h_1 = 4{\cdot}6$; $h_1 \div 3{\cdot}40 = 1{\cdot}354$, $\phi(h_1 \div 3{\cdot}40) = 1{\cdot}245$, and the left-hand side of the equation equals $+ {\cdot}17$.

Plotting the three values $- {\cdot}94$, $+ {\cdot}17$, and $+ {\cdot}77$ against the corresponding values of h_1, and drawing a smooth curve through them, this curve intersects the zero axis where $h_1 = 4{\cdot}52$ ft., showing that this value of h_1 is the solution of the equation.

The accurate solution of this particular case has been worked out by Jameson,* who has plotted the surface profile. At this point, the depth as obtained from the profile curve is 4·50 ft., or sensibly identical with that obtained from formula (1), p. 67. This case has been considered in detail, because the circular section diverges at least as far from the conditions assumed in obtaining this formula as does any section likely to be adopted in practice.

Where the width or depth of the channel varies appreciably, the problem may be investigated by taking the channel in stages, in each of which the variation is relatively small.

* A. H. Jameson, M.Inst.C.E., Institution of Water Engineers, 5th December, 1919.

46. Flow over Weirs.—The flow over a weir can be expressed as

$$Q = KbH^{\frac{3}{2}} \text{ c. ft. per second}$$

where b is the length of the weir in feet;
where H is the head over the crest, measured to the level of still water behind the weir;
where K is an experimental coefficient, which varies with the type and conditions of discharge.

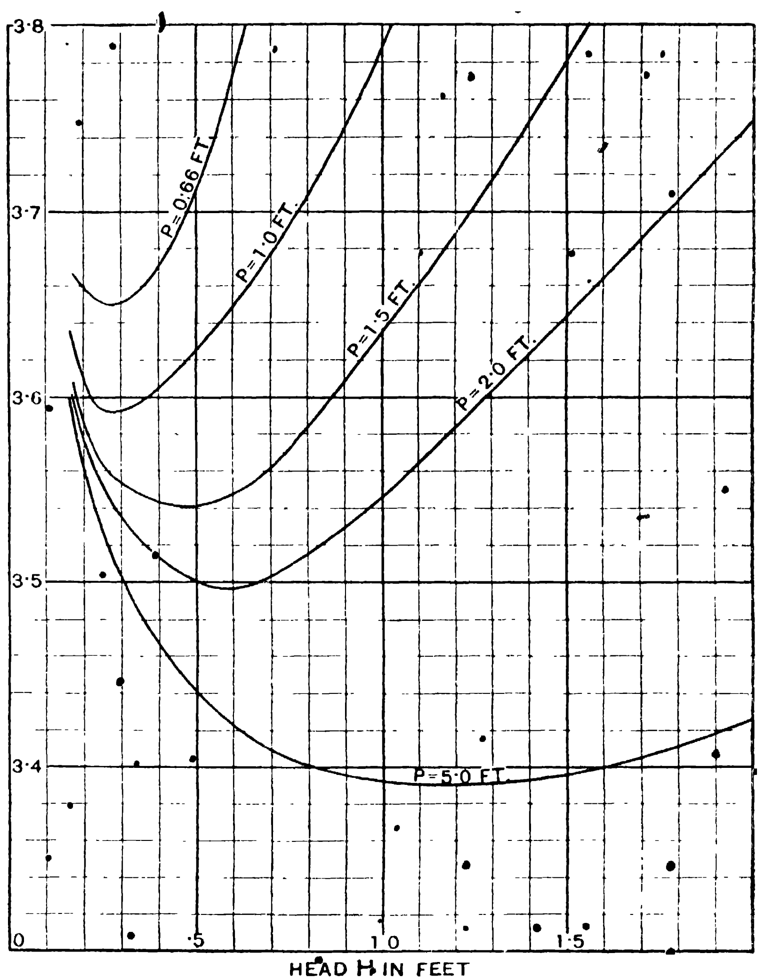


Fig. 34.—Values of K in Bazin's Weir Formula.

Sharp-edged Rectangular Weirs.—In the case of a rectangular weir having a thin sharp-edged crest and a vertical up-stream face, the two most useful formulæ are those of Francis and of Bazin.

In the Francis formula $K = 3{\cdot}33$, while b is replaced by $b - 0{\cdot}1nH$,

where n is the number of full end contractions. A weir with no end contractions is said to be "suppressed". In the Bazin formula,

$$K = \left(3·25 + \frac{·0789}{H}\right)$$

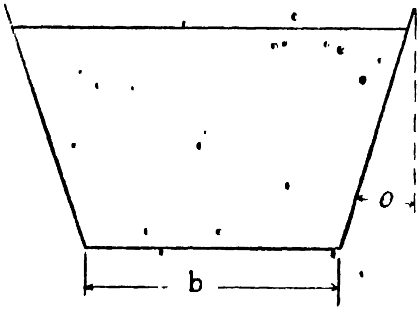


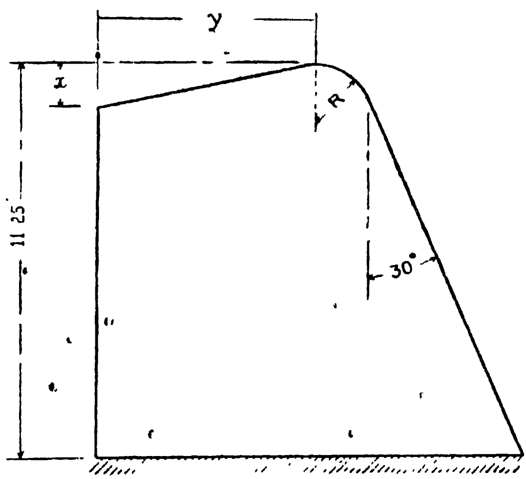
Fig. 35.—Cipoletti Weir

for a suppressed weir. These values of K apply where the area of the approach channel is so relatively large that the effect of the velocity of approach may be neglected. If, as is usually the case in a suppressed weir, the velocity of approach is appreciable, the formulæ become:

Francis, $Q = 3·33(b - 0·1nH)\{(H + h)^{\frac{3}{2}} - h^{\frac{3}{2}}\}$ c.f.s.

Bazin, $Q = \left\{1 + ·55\left(\frac{H}{P + H}\right)^2\right\}\left\{3·25 + \frac{·0789}{H}\right\}bH^{\frac{3}{2}}$ c.f.s.,

where, in the Francis formula, $h = v^2 \div 2g$, and where v is the velocity of approach, while, in the Bazin formula, P is the height of the weir crest above the bed of the approach channel.



Weir	x	y	R
A	1·0	6·0	1·0
B	·95	3·66	3·0
C	·75	3·0	3·0
D	1·5	3·0	3·0
E	3·0	3·0	3·0

Fig. 36

The curves of fig. 34 show values of K corresponding to different heads and depths of approach channel, as deduced from Bazin's results.

The above formulæ apply only to a weir having free access of air to the under side of the falling sheet or nappe. If the nappe clings to the crest or front face of the weir, or if free access of air is prevented, the discharge is increased.

Triangular Weirs.—If the weir is thin-crested and sharp-edged, and if θ be the angle between its two sides,

$$Q = 4{\cdot}28\, c \tan\frac{\theta}{2} \cdot H^{\frac{5}{2}} \text{ c.f.s.,}$$

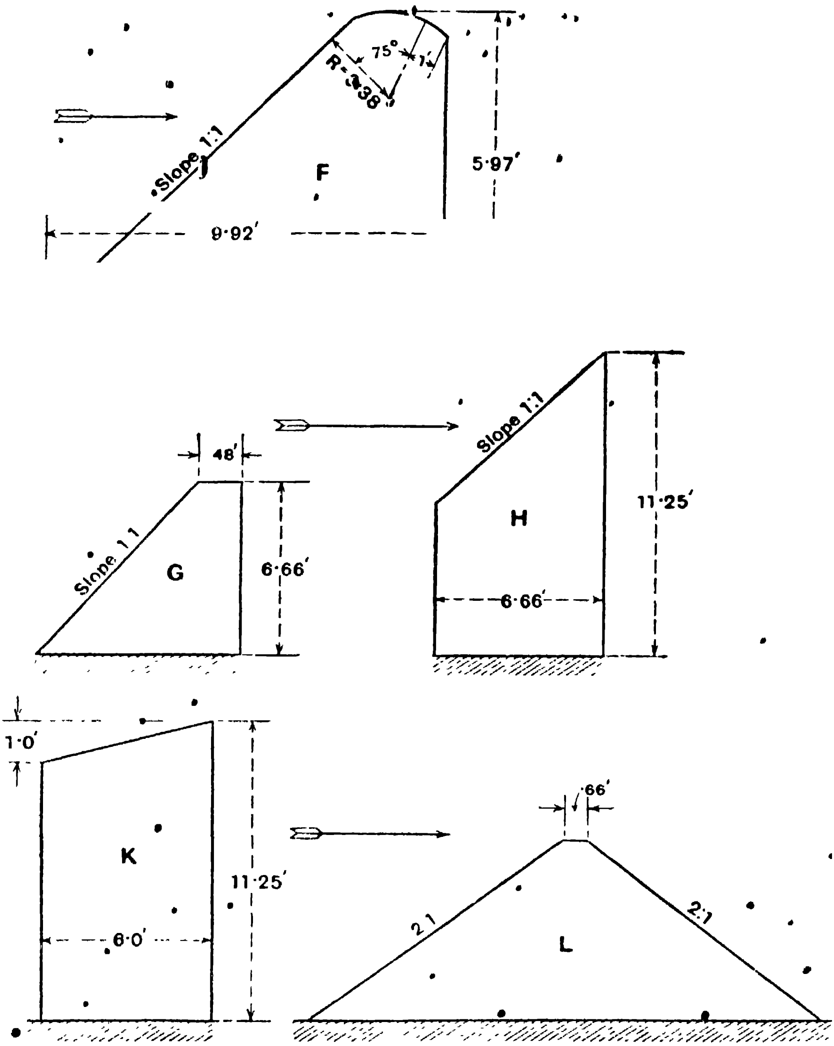


Fig. 37.—Types of Broad-crested Weir

where c depends slightly on θ, and H is measured in feet.

If $\theta = 90^\circ$, $c = {\cdot}593$, and $Q = 2{\cdot}536\, H^{\frac{5}{2}}$ c.f.s.

If $\tan\frac{\theta}{2} = 2$, $c = {\cdot}618$, and $Q = 5{\cdot}29\, H^{\frac{5}{2}}$ c.f.s.

Cippoletti Weir.—If the sides of a weir having two end contractions be inclined outwards at an angle θ with the vertical (fig. 35), the value of K

in the formula $Q = KbH^{\frac{3}{2}}$ is sensibly independent of the head if θ is such that the side slope is 1 horizontal to 4 vertical. Such a weir is called a *Cippoletti weir*. The discharge is given by

$$Q = 3{\cdot}37\, bH^{\frac{3}{2}} \text{ c.f.s.},$$

if the velocity of flow in the approach channel is negligible, and by

$$Q = 3{\cdot}37\, b\{(H + h)^{\frac{3}{2}} - (h)^{\frac{3}{2}}\} \text{ c.f.s.},$$

as in the Francis formula, when the velocity of approach is taken into account.

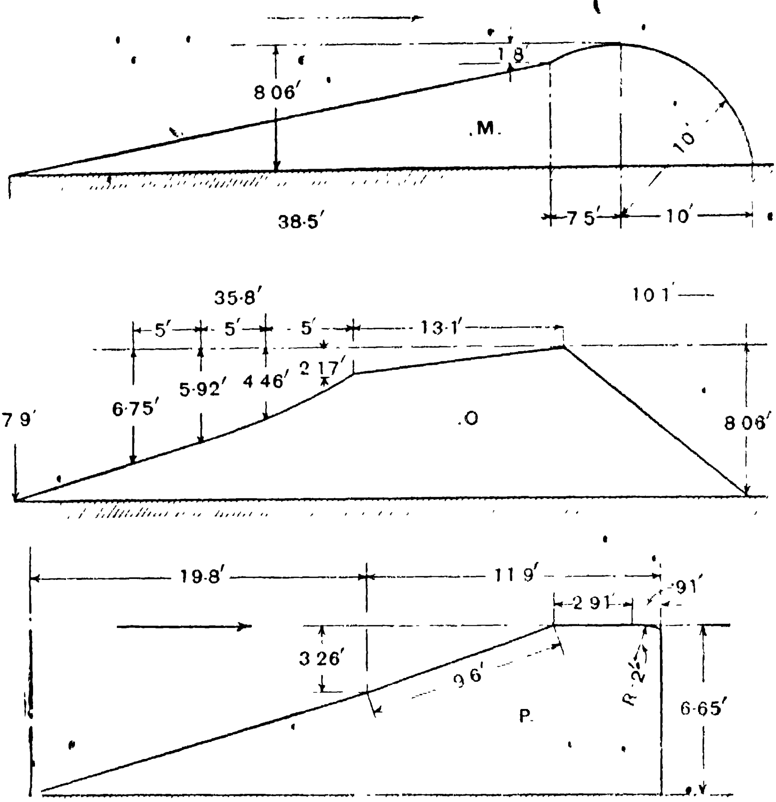


Fig. 38.—Types of Broad crested Weir

Broad-crested Weirs.—Experiments indicate that if the width of the crest of a sharp-edged weir is less than about ·33 H, the nappe will spring clear of the crest. Weirs with wider crests, in which the nappe adheres to the crest, are termed broad-crested weirs. Expressing the discharge over such a weir as

$$Q = K^1 bH^{\frac{3}{2}},$$

the value of K^1 has been determined* for the case of a weir having a sharp

* See U. S. Geological Survey: *Water Supply and Irrigation*, Paper No. 200, R. E. Horton. Also "Rafter", *Trans. Am. Soc. C. E.*, Vol. 44, 1900.

upper edge and a horizontal flat crest, and for weirs of the forms shown in figs. 36 to 38.

These values are as follow:

Flat-crested Weir

H (feet).	Width of Crest. 0·48 ft.	0·93 ft.	1·65 ft.	3·17 ft.	5·89 ft.	8·98 ft.	12·24 ft.	16·30 ft.
0·5	3·01	2·76	2·73	2·66	2·61	2·61	2·61	2·61
1·0	3·24	3·01	2·93	2·70	2·67	2·66	2·65	2·64
1·5	3·33	3·19	3·03	2·73	2·69	2·67	2·67	2·65
2·0	3·33	3·29	3·08	2·73	2·68	2·67	2·66	2·65
3·0	3·33	3·33	3·12	2·71	2·65	2·64	2·62	2·61
4·0	3·33	3·33	3·15	2·69	2·63	2·61	2·60	2·59

Weirs (A to F)

H (feet)	A.	B.	C.	D.	E.	F.
0·5	3·21	3·10	3·22	3·23	3·23	3·23
1·0	3·42	3·27	3·35	3·46	3·46	3·27
1·5	3·54	3·38	3·44	3·61	3·64	3·40
2·0	3·55	3·44	3·47	3·68	3·75	3·46
3·0	3·30	3·48	3·48	3·75	3·87	3·87
4·0	3·14	3·51	3·48	3·81	3·96	3·65

Type G

Up-stream slope.	1 to 1.	2 to 1.		3 to 1.	4 to 1	5 to 1.
Crest width (ft.).	·48.	·33*.	·66*.	·66*.	·66*.	·66.*
Head in feet. 0·5	3·22	3·35	3·22	3·64	-	3·31
1·0	3·57	3·68	3·44	3·82	3·44	3·33
1·5	3·59	3·83	3·59	3·83	3·46	3·34
2·0	3·60	3·77	3·66	3·69	3·48	3·35
3·0	3·58	3·68	3·68	3·55	3·48	3·38
4·0	3·55	3·70	3·70	3·55	3·48	3·39

Types H to P

Head (feet).	H.	K.	L.	M.	N.†	O.	P.
·5	3·53	3·47	3·14	3·21	2·91	3·65	3·06
1·0	3·59	3·46	3·42	3·21	3·16	3·63	3·05
1·5	3·64	3·45	3·52	3·20	3·33	3·61	3·04
2·0	3·65	3·42	3·61	3·16	3·42	3·56	3·11
3·0	3·63	3·35	3·66	3·06	3·51	3·45	3·20
4·0	3·61	3·29	3·66	3·01	3·58	3·38	3·27

* "Rafter". Crest height, 4·7 ft.

† Type N is the same as M, but with the addition of a 12-in. square baulk to the crest.

47. Precautions to be adopted in Weir Gaugings.—The standard sharp-edged weir having a free discharge, or, for small quantities, the right-angled triangular notch, are the only types for which the coefficients have been determined with sufficient accuracy to admit of use for accurate measurement of flow without previous calibration.

For accurate measurement the following are essentials:

1. Sharp-edged weir sill, fixed so as to be incapable of vibration, having its face vertical and perpendicular to the direction of the stream, and, if rectangular, having its sill horizontal.

2. Clear discharge into air, with no adherence of the vein to the weir face.

3. Weir long in proportion to its depth, i.e. $b > 3\,H$.

4. H small in comparison with the depth of the approach channel, and sectional area of vein ($b\,H$) not greater than one-sixth that of this channel.

5. Suitable channel of approach. This should be as long and of as uniform section as possible so as to allow of the motion becoming steady before reaching the weir. The length should, if possible, exceed 30 H, this ratio being increased where the length of weir is largely in excess of 3 H. In Bazin's experiments the length of supply channel was 49·2 ft. with a maximum head of 1·97 ft. and a maximum weir length of 6·56 ft., giving a length = 25 H.

6. Accurate determination of the head H. For accurate work the surface-level should not be taken in the stream itself, but in a stilling box or pit from 18 in. to 2 ft. square communicating with the stream through a pipe of about 1 in. diameter. The zero of the gauge should be accurately adjusted to the level of the weir crest. This may conveniently be done by driving a post into the bed of the stream above the weir, until its upper end is above the level of the crest. A brass peg is driven into this and is filed down until, as shown by straight-edge and level, its point is exactly level with the crest. The water-level may then be adjusted with great accuracy until this point is in the surface, when the zero level may be read off on the gauge. Where it is impossible to bypass the flow past the weir for this preliminary work, a vertical post fixed near the weir on its down-stream side may be graduated by straight-edge and level, to give heights above the crest. These may then be transferred by straight-edge and level to a graduated scale in the measuring pit, or may be used to give the datum level to which to adjust the zero reading of the hook or float gauge. For accurate work, where individual readings are to be taken, a hook gauge (fig. 14*a*), provided with a vernier for reading to the nearest ·001 ft., and with screw adjustment, is best. Where an automatic record is to be kept, the level is recorded by a float operating the recording pencil. For rough work the level may be taken on a vertical staff gauge driven into the bed of the stream or attached to some convenient support. Such a gauge should be surrounded by a stilling box in order to damp out surface oscillations.

48. Time required to lower the Level in a Reservoir, by discharge over a Weir.—Since the volume discharged by the weir is $Kbh^{\frac{3}{2}}$, the rate of fall of the surface-level will be $Kbh^{\frac{3}{2}} \div A$, or

$$\frac{dh}{dt} \qquad Kbh^{\frac{3}{2}}$$

Integrating this, the time ($t_2 - t_1$ sec.) required to reduce the head on the weir from h_1 to h_2 ft., is given by

$$t_2 - t_1 = T = \frac{2A}{Kb}\left(\frac{1}{\sqrt{h_2}} - \frac{1}{\sqrt{h_1}}\right) \text{ sec.}$$

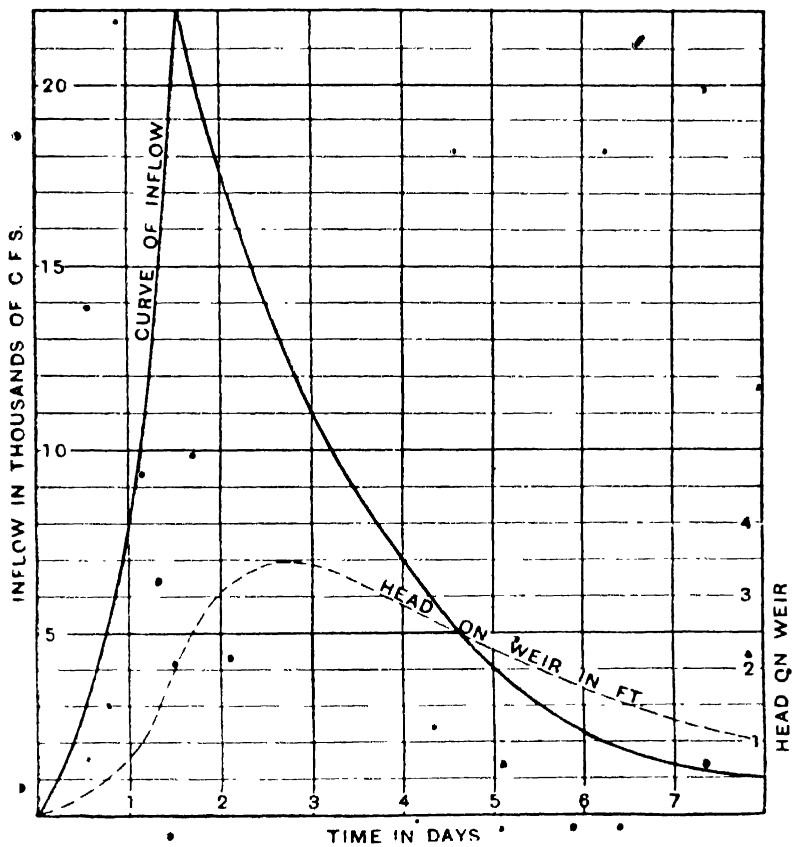


Fig. 39. Flood Curve for Harper River, New Zealand

49. Time to raise Level during Flood Discharge.—Since a flood may occur when a storage reservoir is already full, provision must be made for discharging flood water without an abnormal increase in height behind the dam, and it becomes important to be able to determine the rate at which the surface-level rises, and at which it falls as the flood subsides.

Floods always rise rapidly to a maximum and subside more slowly. Fig. 39* shows the rise and fall of a typical flood in the Harper River, New Zealand, and also shows the corresponding change of level in a reservoir of 400×10^6 sq. ft. surface area, discharging over a spillway weir 660 ft. long with a weir coefficient of 3·0, and subject to this inflow.

If Q be the rate of inflow at a given instant, and H the corresponding head on the weir, the excess of inflow over outflow will be

$$Q - Kbh^{\frac{3}{2}} \text{ c.f.s.,}$$

and the rate at which the surface-level is rising is given by

$$\frac{dh}{dt} = \frac{Q - Kbh^{\frac{3}{2}}}{A} \text{ f.s.}$$

This can be integrated if A and Q are constants, and writing $Q = KbH^{\frac{3}{2}}$, where H is the head over the weir when the discharge is Q and $r = h \div H$, we get

$$T = \frac{2A}{3Kb\sqrt{H}}\left[\log_e \frac{\sqrt{1+r+\sqrt{r}}}{1-\sqrt{r}} - \sqrt{3}\left\{\tan^{-1}\frac{2}{\sqrt{3}}\left(\sqrt{r}+\frac{1}{2}\right) - \frac{\pi}{6}\right\}\right]$$

$$= K'\phi(r), \text{ where } K' = 2A \div 3Kb\sqrt{H}.$$

Values of $\phi(r)$ have been tabulated by Gould † and are given below.

Values of Gould's Function $\phi(r)$.

r.	·00.	·01.	·02.	·03.	·04.	·05.	·06.	·07.	·08.	·09.
0	·0000	·0153	·0306	·0459	·0613	·0766	·0919	·1072	·1226	·1378
1	·1532	·1685	·1838	·1992	·2155	·2319	·2483	·2646	·2810	·2973
2	·3137	·3301	·3464	·3628	·3791	·3955	·4137	·4319	·4501	·4683
3	·4865	·5047	·5229	·5411	·5593	·5775	·5957	·6139	·6321	·6534
4	·6747	·6960	·7173	·7386	·7598	·7811	·8024	·8237	·8450	·8663
5	·8876	·9137	·9399	·9660	·9921	1·018	1·044	1·070	1·097	1·113
6	1·149	1·175	1·201	1·232	1·267	1·303	1·338	1·373	1·409	1·444
7	1·479	1·514	1·550	1·585	1·620	1·656	1·707	1·759	1·811	1·862
8	1·914	1·966	2·018	2·071	2·149	2·226	2·303	2·381	2·458	2·535
9	2·613	2·768	2·923	3·078	3·235	3·389	3·544	4·010	4·475	4·940

The above equation gives the time necessary for the level to rise from $h = o$ to $h = r$H. If the rate of variation of Q with time is known, this may be taken as constant over a small range of surface-level, say ·5 ft. For example, for the first half-foot the time is given by

$$T_1 - T_0 = K'\left\{\phi\left(\frac{\cdot 5}{H}\right) - o\right\}.$$

Now with the new Q, and, if necessary, the new value of A and of K,

* E. Parry, B.Sc., M.I.E.E.; *Impounding Flood Water*, P.W.D., Wellington, 1919. This paper gives a graphical method of determining the rate of rise and fall of the surface-level in the storage reservoir. † *Engineering News*, 5th December, 1901.

calculate the new value of K′, say K″. Then the time for the level to rise from ·5 ft. to 1·0 ft. is given by

$$T_2 - T_1 = K''\left\{\phi\left(\frac{1}{H}\right) - \phi\left(\frac{\cdot 5}{H}\right)\right\},$$

and so on.

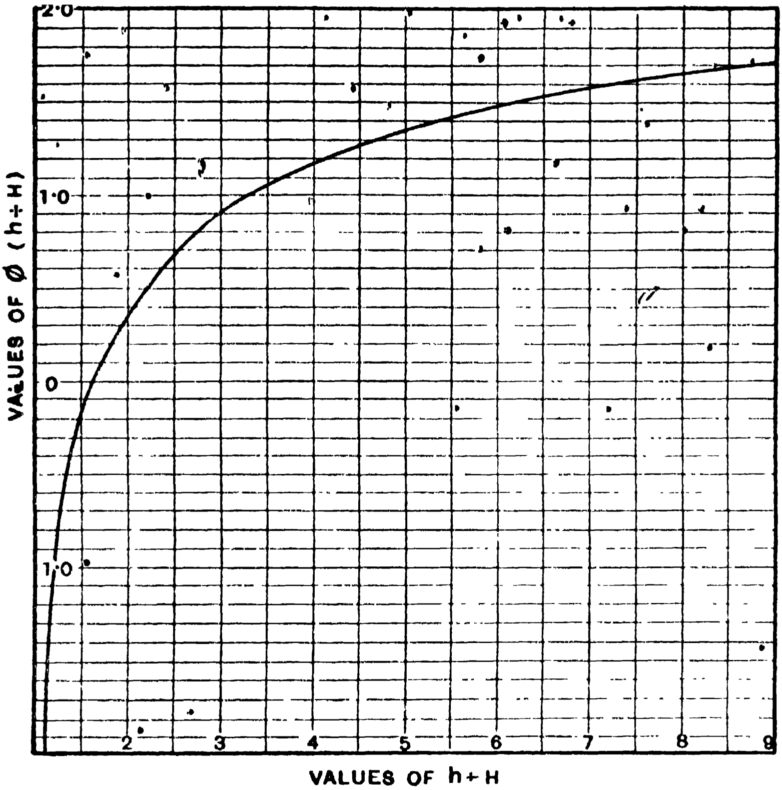


Fig. 40.—Values of $\phi(r)$ for Lowering of Surface Level of a Reservoir by Discharge over a Weir

If the rate of inflow is less than the discharge, so that the surface-level is falling, $Q - KbH^{\frac{3}{2}}$ is negative, and the solution of the equation becomes

$$T_2 - T_1 = \frac{2A}{3Kb\sqrt{H}}\left\{\phi\left(\frac{h_1}{H}\right) - \phi\left(\frac{h_2}{H}\right)\right\},$$

where $\phi\left(\frac{h}{H}\right)$ or $\phi(r)$ has the value

$$\log_e \frac{\sqrt{r} - 1}{\sqrt{1 + r + \sqrt{r}}} + \sqrt{3}\tan^{-1}\frac{2\sqrt{r} + 1}{\sqrt{3}},$$

r now being greater than unity. This gives the time to lower the surface-level from h_1 above the weir crest, to h_2. Values of this $\phi(r)$ are given in the graph of fig. 40.*

* For further information on this subject an article by R. E. Horton, " Determining the Regulating Effect of a Storage Reservoir ", *Engineering News Record*, 5th September, 1918, p. 455 may be consulted.

Reservoir of Varying Cross-sectional Area.—In the foregoing analyses, the area of the reservoir has been assumed to be sensibly independent of the depth of water over the range of depths occurring during the period of overflow. Where this is not true, and where the area corresponding to a given depth is known, the foregoing equations are best solved by graphical integration.

CHAPTER VI

The Development of Water-power Schemes

Classification of water powers; schemes of development; power available; definition of head.

50. Classification of Water Powers.—Hydraulic schemes may be roughly classified according as they utilize a high, medium, or low head. While there is no definite line of demarcation between a high and a medium head, or between a medium and a low head, a high-head installation is usually understood to be one using a head between 500 and 5000 ft., and developing its power from Pelton wheels. A low-head plant is one in which reaction turbines, usually submerged, are used, and in which the head is anything between the lowest practicable limit of about 1·5 ft. and 80 ft. Medium heads lie between these limits, and use Pelton wheels or reaction turbines, depending on the volume of water available and on local circumstances. More recently the reaction turbine has been used to an increasing extent in the region of high heads formerly thought suitable only for the Pelton wheel, and the limiting head up to which either type may be used is at present in the neighbourhood of 750 ft.

51. Schemes of Development.—The working head may be obtained in a variety of ways.

1. From a natural waterfall, with a pipe line conveying the water from above the fall to a power house situated on the bank of the stream below the fall. This may be either a high-, medium-, or low-fall installation.

2. By a dam erected across a stream, the head being due solely to the dam. The head is comparatively low, and a direct development is usually possible. Four types of direct development are illustrated in the sketches of fig. 41. In A and B the power house is placed at or near the end of the spillway. In C it is placed on the dam itself. This is only possible when the remaining length of spillway is sufficient to take the flood discharge without an excessive increase in the head behind the dam. In a river with rocky, steep, and narrow banks, the entire width of the channel must be available for a spillway, and, owing to the necessity for rock excavation the construction of a power house at the end of the dam may be very costly. In such a case a hollow ferro-concrete dam, with its interior

utilized for the power house, offers an excellent solution. The whole of the crest is then available as a spillway (figs. 41*d* and 51).

3. From a natural or artificial fall by tapping a stream or reservoir and conveying the water in an artificial channel to a point from which

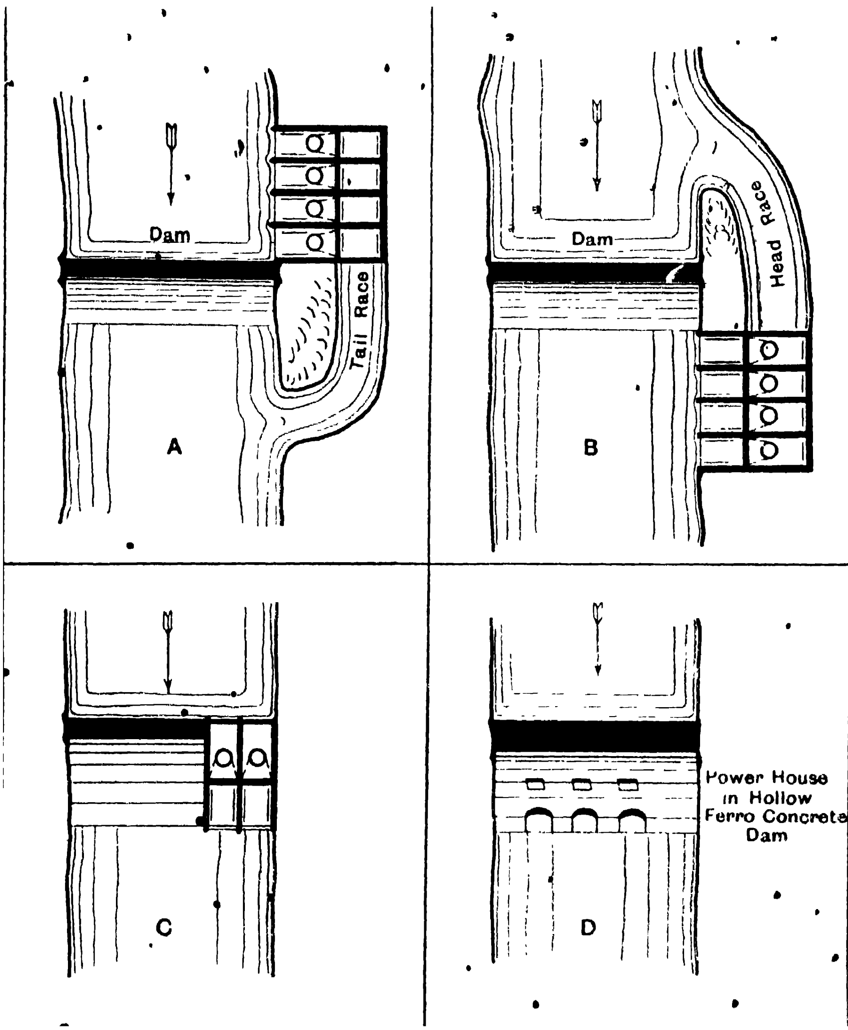


Fig. 41.—Types of Low-head Development.

a pipe line can be taken to a power house at a lower level. This method is applicable to any head. The channel, which may consist of an excavated canal, or a wooden, steel, or concrete flume, follows a contour of the hill-side at a slight gradient of from 1 in 500 to 1 in 1500 (fig. 42*a*). In some cases the stream makes a hairpin bend, and the water may be conveyed in a tunnel driven through the intervening spur. Such a tunnel is usually at

head-water level, but may, if desirable, be driven at tail-water level and act as the tail race (fig. 42 *b* and *c*). This method is not in general

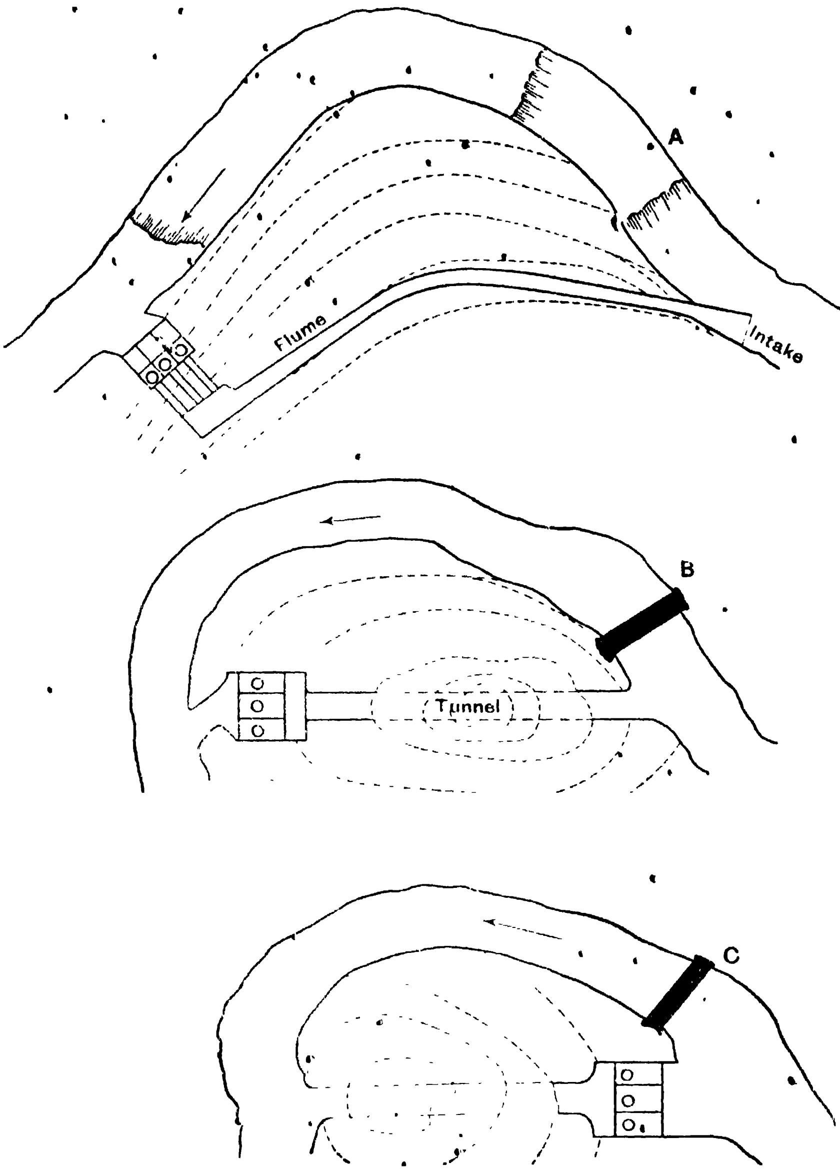


Fig. 42.—Types of Water-power Development

to be advocated. In the case of a high-head development it means installing the turbines at the bottom of a well sunk to tail-race level, and forms a costly and inconvenient scheme. In the case of a low-head development this might not be necessary, but even then the method is

only of advantage where the value of the ground over which a head race would require to be constructed is very large.

Where water is available in an elevated valley sloping gently to the plains in one direction, and where, on the other side of the watershed, the slope is steep, the lower end of the valley may be dammed and the water may be diverted from its natural watershed by means of a tunnel driven through the intervening ridge, and may be piped down to a power station at the foot of the steeper slope on the other side of the ridge. Such an arrangement is indicated in fig. 43.*

The scheme to be adopted depends essentially on the physical characteristics of the site, and should be such as to utilize the maximum possible proportion of the head at the least cost, and at the same time to minimize as far as possible any interruptions due to floods or droughts. While the existing

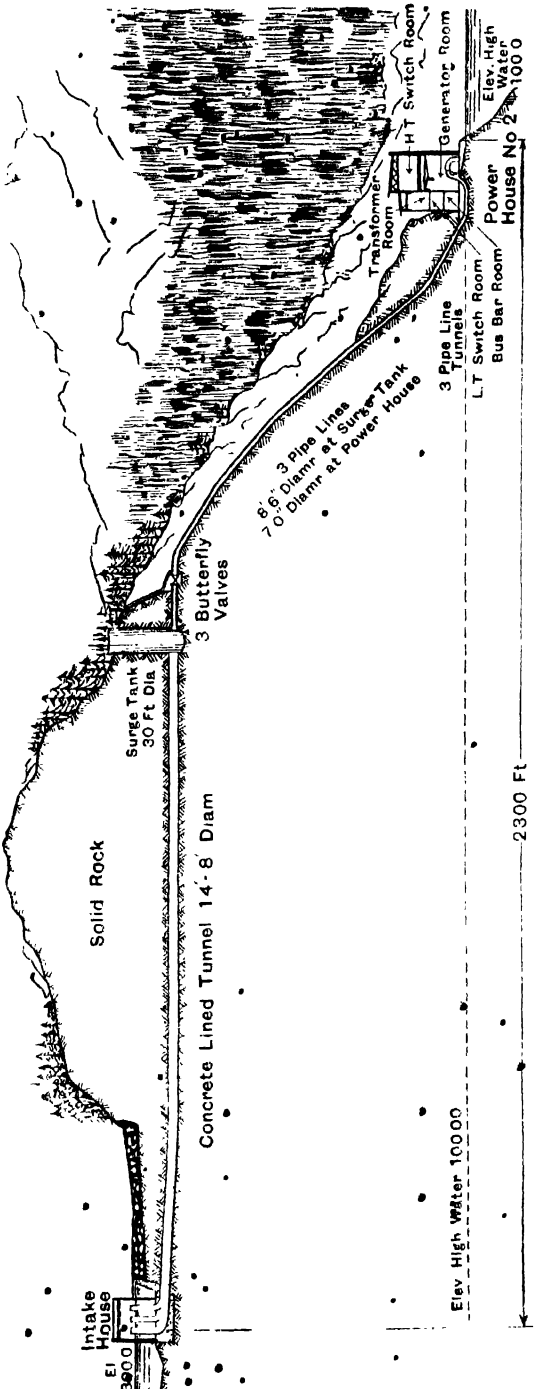


Fig. 43.—Types of Water-power Development involving use of Pressure Tunnel, Surge Tank and Pipe Line

* Department of the Interior, Ottawa: adapted from *Water Resources*, Paper No. 13, 1915.

conditions usually point to one or other scheme as being most suitable, in some cases the relative advantages of more than one type of development require to be carefully investigated.

52. Power available from a Given Fall.—The theoretical horse-power or water horse-power available from a supply of Q c.f.s. under a head of H ft. is equal to

$$\frac{62{\cdot}4\ QH}{550}$$

With a turbine whose efficiency is η, the brake horse-power at the turbine shaft is

$$\frac{62{\cdot}4\ QH\eta}{550}.$$

While turbine efficiencies exceeding 90 per cent are on record, the efficiency of the average turbine is more nearly 80 per cent, and adopting this value the brake horse-power becomes $QH \div 11$. If the efficiency of the electric generator is η', the electric horse-power delivered at the switch-board mains is η' times the brake horse-power at the turbine shaft. In general η' is about ·95, so that the e.h.p. is approximately $QH \div 11{\cdot}6$, and since 1 kilowatt-hour is equivalent to 1·34 horse-power-hours, the electrical output will be approximately $QH \div 15{\cdot}5$ kw.

53. Definition of Head in a Power Plant.—The total head in a hydraulic installation is the difference in level between the water in the supply reservoir and in the tail race. Where it is necessary to convey the water to the turbines through a canal or pipe line, the effective head at the turbines is less than the total head, by the head lost in friction and eddy formation in the water conductor.

In computing the over-all efficiency of the plant, the total head is to be used, and where the velocities of flow in the intake ahead of the racks and in the tail race are such as to make the velocity heads at these points appreciable, the equivalent velocity head should be added to each measured level in computing the total head. The value of the velocity head is to be taken as $v^2 \div 2g$ ft., where v is the mean velocity in feet per second over the section in question.

In computing the efficiency of the turbine, the losses between the intake and the turbine should not be debited to the turbine, nor should the head equivalent to the velocity of discharge from the end of the draft tube. In the case of a low-head turbine this latter head may be relatively important. Thus with a discharge velocity of 4 f.s., the velocity head $v^2 \div 2g = 0{\cdot}25$ ft., and if no account were taken of this, it would make a difference of 2·5 per cent in the apparent efficiency in the case of a turbine operating under a head of 10 ft. With a cased turbine the effective head is the difference between the sum of the pressure, potential, and velocity heads at the entrance to the turbine casing, and the sum of the same heads at the point of exit from the draft tube, levels in each case being measured

to the centre of the pipe or draft tube. The pressure head at the centre of a submerged draft tube is to be taken as equal to the depth of this point below the highest level of the water in the tail race.

In the case of a Pelton wheel the effective head is to be taken as the sum of the pressure, potential, and velocity heads in the pipe line behind the turbine nozzle, the level of the centre of the nozzle being taken as datum.

CHAPTER VII

Civil Engineering Problems

Dams and headworks; dam accessories; gates and valves; intakes; flumes, tunnels, and pipe lines; pipe-line accessories; tail races.

54. In most hydro-electric schemes a dam is required to hold up the water, either to create or increase the working head, to create storage, or to assist in forming the intake from which water is taken to the turbines.

There are four main types of dam, which are differentiated according to the method by which stability against the pressure of the water is secured.

(1) Gravity or solid dams, in which the weight of the dam itself provides the stability. In these the up-stream face is vertical or nearly so.

(2) Buttressed or hollow dams. These have the up-stream face inclined at an angle of about 45°, so that the water pressure on it has a considerable vertical component, and thus provides the stability.

(3) Single-arch dams. These are built in the form of a circular arch in plan, the thrust of the water being transferred to the sides of the valley.

(4) Multiple-arch dams. Instead of a single arch, several arches are used, buttresses being used to convey the down-stream thrust of the water to the foundations, the side thrust being transmitted to the valley sides.

These types grade into one another. Many dams of the solid gravity section are curved in plan, and gain additional stability from the arch action, while multiple-arch dams are usually built with inclined faces, so that they are really buttressed dams, with curved decking.

55. Gravity Dams. These may be constructed of timber crib, earth, rock fill, or masonry. Local considerations govern the choice of type. It is not advisable to use an earthen or rock-fill dam on a rock or shale foundation owing to the difficulty of getting a watertight joint between the two dissimilar materials. On such a foundation a masonry or concrete dam is to be used. Where no rock foundation is available one of the other types is usually preferable, and the choice then depends largely on the availability of suitable constructional materials.

Timber-crib Dams may be used for low heads up to about 30 ft., where timber is cheap. They consist of a framework of logs bolted or

spiked together, and weighted with rock or gravel. A plank decking is provided on the up-stream side to prevent leakage, and, if the dam is required to act as a spillway, also on the down-stream side (fig. 44). Such a dam may be used on either a rock foundation or on soft ground. In the former case the framework is bolted down to the rock. In the latter case a wider base is required, and an apron must be extended for a sufficient distance down stream to prevent scouring of the foundation due to any overflow. Such dams are usually only used for temporary work.

56. Earthen Dams have been used for heads as high as 200 ft., but so great a height is very unusual. On no account is water to be allowed to flow over the crest, and if it is necessary to discharge flood waters over the dam, a masonry spillway, at a lower level than the crest of the dam, must be provided. It is often preferable to build an independent spillway

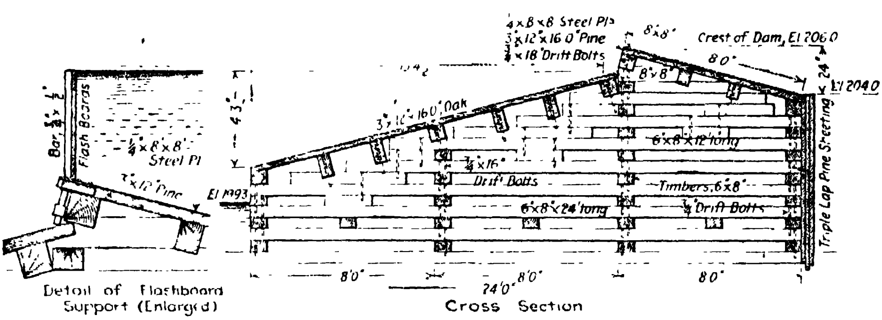


Fig. 44. Timber Crib Dam

discharging through a tunnel excavated around the flank of the dam as shown in fig. 45. In general the width of the top of the dam is from one-third to one-fourth of the height. The up-stream face has a slope of about 3 to 1, and the down-stream face about 2 to 1.

Any appreciable percolation of water through such a dam or its foundations is dangerous, as leading to ultimate disintegration, and, in order to prevent this, a central trench is usually sunk into an impermeable stratum, and a watertight wall of puddled clay, or clay and gravel, is built in this trench and extended above the water-level. The bottom width of this wall should be about one-third the height and the top width one-sixth the height. This wall is supported by the earth filling, which should be put in place with the most impervious material near the centre of the dam (fig. 46). Where suitable material for this wall is difficult to obtain, it may be replaced by a thin concrete or masonry cut-off wall, or by a wall extending only to the original ground surface. The earth over the whole area covered by the dam must be removed to a depth of one or two feet, and all tree stumps, &c., extracted. The exact type of construction depends largely on the character of the material available near the dam site. Where this consists of a suitable mixture of gravel, sand, and clay, the puddle wall may be eliminated. The most suitable mixture

consists of about 60 per cent coarse gravel, 20 per cent fine gravel, 8 per cent sand, and 12 per cent clay. The sand and gravel give the necessary stability, and the clay the necessary cohesion.

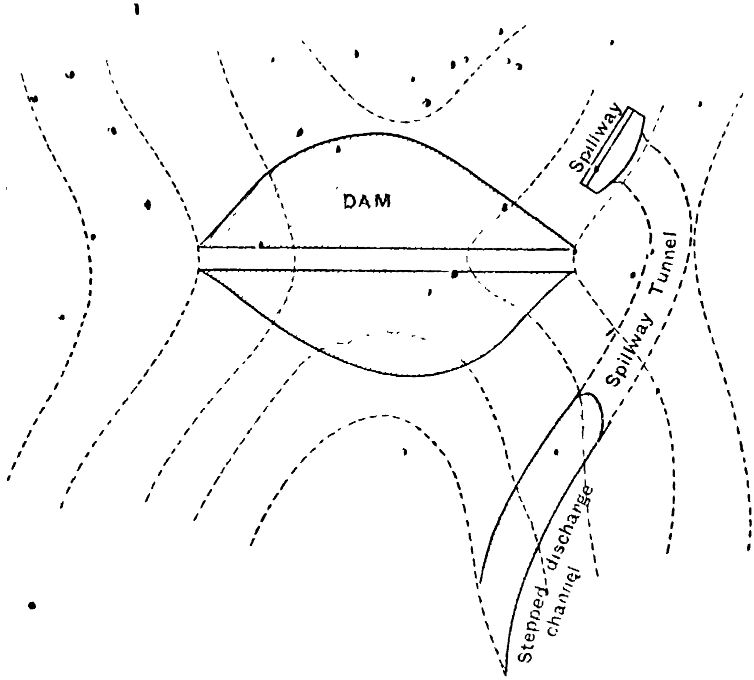


Fig 45 Arrangement of Spillway for Earthen Dam

The materials may be placed in position either by dumping from trestles or cableways, or where water is available and the material is at a

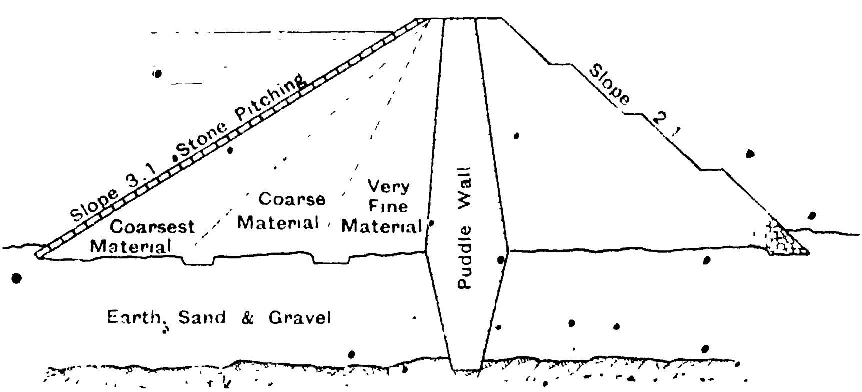


Fig 46 Earthen Dam Section

higher elevation than the dam, by the hydraulic fill method. Here the material is sluiced by a high-velocity jet of water into a flume which conveys it on to the dam, where it is deposited as the water drains away. This

gives a very compact and impervious structure. When deposited by dumping, it should be put down in layers not exceeding 12 in. thick, each layer being carefully rammed before the succeeding layer is put down.

Where feasible the outlet pipes or conduit should be taken through the undisturbed ground on one flank of the dam, and not through the dam itself, on account of the danger of percolation along the outside of the pipes, and of uneven settlement along the line of the pipes.

The up-stream face must be protected against erosion either by a concrete face or by rip-rap. The down-stream face may be protected in the same way, but is more generally sown with grass seed. In a high dam, the down-stream face should be terraced, so as to reduce the velocity of any water flowing down its slope. Care must be taken to provide adequate drainage from the down-stream side of the core wall, where this is incorporated in the design.

Rock-fill Dams are simply earthen dams in which the down-stream portion is composed of rock filling which supports the impervious earth filling. In this case the down-stream slope is usually about $1\frac{1}{2}$ to 1.

57. Masonry Dams may be constructed of coursed ashlar, of rubble or cyclopean masonry, or of concrete.

Stability.—The forces acting on any section of a masonry dam exposed to water pressure on one face are the weight of the section acting vertically downwards, and the pressure of the water everywhere normal to its face. The dam may yield

(1) by overturning about any horizontal joint;
(2) by sliding of one horizontal section over that below it;
(3) by crushing of the foundations or of any joint, due to excessive compressive stress;
(4) by shearing of the material in the plane of maximum shear.

Consider a section of the dam, of unit width (fig. 47). The resultant, R, of the forces P and W acting on the section above any joint FE, and also its line of action OR, may be determined by graphical composition of forces as indicated in the figure. Here the force W, equal to the weight of masonry above the joint EF, acts vertically through G, the centre of gravity of the mass. The water pressure P is normal to the surface, and has a value of $32{\cdot}2h^2$ lb., where h is the depth of the joint. It acts through the centre of pressure of the area from Q to E, i.e. at a point $\frac{2}{3}$ QE from Q. The resultant R therefore passes through O, the point of intersection of the lines of action of P and W, and if the line OW represents the magnitude of the weight W, the line OR will represent R to the same scale. Since OR cuts FE at C, this is the centre of pressure on the joint, and, in order that no portion of the joint should be in tension, should lie within its middle third. The only condition necessary in order that overturning should not take place is that the centre of pressure should lie somewhere within the joint, so that a joint in which it lies within the middle third cannot yield by overturning. For the dam shown, HCP marks the line of

action of the resultant forces at each point, and since the lines ALK, BDN mark the boundaries of the middle thirds, the first condition for stability is fulfilled. Lines of pressure should be drawn in this way for the dam, both full and empty, and both lines should lie within the middle thirds

The resultant pressure on any horizontal section may be resolved into a normal and a tangential component N and T, and if μ be the coefficient of friction at the section, μN must be greater than T if the joint is not to yield by bodily sliding of the upper over the lower portion of the dam, or of the dam over its foundations. For ordinary masonry joints μ may be taken as approximately ·66. Any danger of failure owing to this cause can be removed by inclining the footings so as to bring the direction of the resultant force normal to the joints. Special care should be taken to prevent the percolation of water through the foundations, and to allow for the efficient drainage of their down-stream side. Otherwise a large upward statical pressure may be produced over the footings, which will relieve the footing of a large proportion of the weight, W. The effective value of N is then reduced, and the tendency to bodily sliding of the dam is increased.

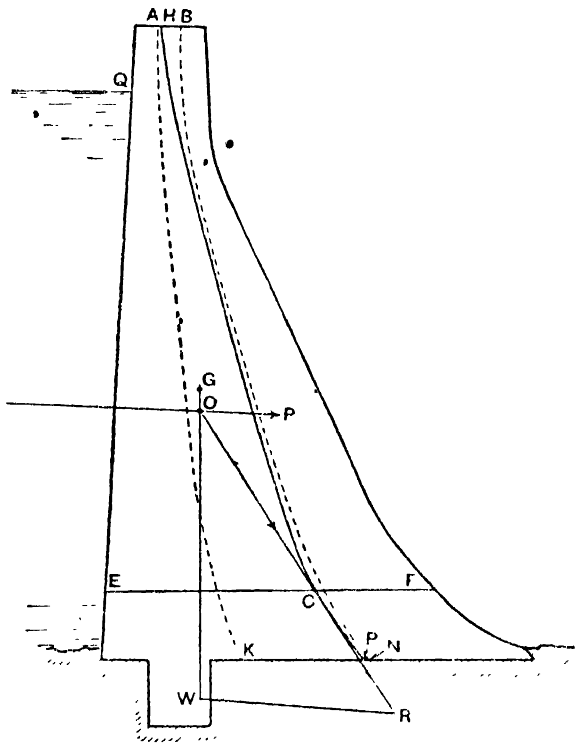


Fig. 47

In practice a trench is often cut in the foundation near the up-stream edge and a concrete cut-off wall built in this trench, while if the foundations are not naturally sound and free from fissures they are made so by cement grouting.

The normal component N is equivalent to a vertical loading which varies uniformly across the joint, and has its maximum value at the down-stream edge of any joint. If b is the breadth of the joint, and x the distance of its centre of pressure from the down-stream edge, this maximum value is equal to

$$p_1 = \frac{2N}{b^2}\left\{2b - 3x\right\},$$

while the minimum value, which occurs at the up-stream edge, is

$$p_2 = \frac{2N}{b^2}\left\{3x - b\right\}.$$

It may be shown that the maximum compressive stress occurs on planes normal to the direction of the resultant, and has a value of $p_1 \div \cos^2\delta$, where δ is the inclination of the resultant to the vertical. For no crushing of the material or the foundations, this value should not exceed the safe bearing stress on the material. If this condition is satisfied, it will be found, in practice, that there is little danger of failure of the material by shearing.

Yielding by crushing is most likely to take place at the toe of the dam. The safe bearing stress depends on the material of the dam and foundations, and may be taken approximately as follows:

Granite or basalt	25 tons per square foot.		
Slates—Hard sandstones	15	,,	,,
Conglomerate	12	,,	,,
Soft sandstones	10	,,	,,
Shale—Hard	8	,,	,,
Shale—Soft	4	,,	,,
Blue clay	3	,,	,,

If the dam is to be used as a spillway, the water pressure due to the head under conditions of maximum overflow is to be used in considerations of stability. Under these conditions the centre of pressure on the front face of the dam will not be at two-thirds the depth, and must be calculated.

Section.—The most economical section for a masonry dam having the water level with the crest would be a triangle having the up-stream face vertical, and the down-stream face sloping at 1 horizontal to $\sqrt{m}$ vertical, where m is the ratio of the specific gravities of the masonry and of water. Thus with limestone, whose weight per cubic foot is 165 lb.,

$$\sqrt{m} = \sqrt{2 \cdot 64} = 1 \cdot 62,$$

and the down-stream slope would be 1 horizontal to 1·62 vertical. In such a dam the stress at the down-stream toe would be zero with the dam empty, and the stress at the up-stream toe would be zero with the dam full.

In practice this elementary section is modified by the necessity for increasing the thickness of the crest to resist ice-pressures or to provide a roadway. For this purpose the upper portion of the dam is made rectangular, as shown in fig. 47, or more generally, for the sake of appearance, the down-stream face is battered out at about 1 in 10. The thickness of the crest is usually made approximately equal to $\sqrt{H}$ in temperate climates, and to $\sqrt{H} + 2$ where ice pressures are feared. The addition of this crest section produces tension in the back of the empty dam at depths greater than $2b'\sqrt{m}$ where b' is the crest width, and to prevent this the front face is battered out at greater depths, at about 1 in 25. In such a dam, of

height H, the maximum stresses occur at the down-stream toe, and are as follow:

$$\text{um compressive stress, } p = w'\text{H}\left(\frac{m+1}{m}\right).$$

$$\text{Maximum sheer stress, } q = w'\text{H}\left(\frac{m+1}{2m}\right).$$

If H is in feet, and w', the weight of masonry, is in pounds per cubic foot, these stresses are in pounds per square foot.

Since the stresses increase directly as H, there will be some limiting height at which the safe crushing stress or bearing stress is attained, and for greater heights it is necessary to splay out the profile as shown in fig. 47, to increase the bearing surface. Thus, assuming a working value of 10 tons per square foot for p, and taking m as 2·25, which corresponds to a value of 140 lb. per cubic foot for w', the limiting height for the triangular section becomes

$$\text{H} = \frac{10 \times 2240 \times 2\cdot25}{140 \times 3\cdot25} = 111 \text{ ft.},$$

Having obtained an approximate section from these considerations, this should be examined by the graphical method of p. 87, and modified if necessary.

Spillways.—In most cases provision must be made to use part of the length of the crest for spilling surplus water, and this provision must be made very ample for dealing with the highest possible flood. The profile of this portion must be carefully designed to give a suitable water path. If the crest is so thin that the water tends to jump clear of the down-stream face, a partial vacuum is created under the nappe, which increases the overturning force on the dam.

The parabolic path of free-spouting water should be set out, and the crest designed so as not to lie below this. The actual curve for construction purposes is often made circular, though a true parabola is preferable.

The toe of the dam must be carried out so as to divert the water away from the foundations. The curve is usually a circle and the radius may be taken as half the head on the dam. The toe should be continued until the curve is horizontal, unless the bed is good rock, when it need not be carried to quite this extent. The extension of the toe beyond the normal profile of the down-stream face is usually disregarded in calculating the stability.

Temperature Effects.—Straight dams are apt to develop cracks due to contractions in cold weather, and in a dam of moderate thickness such cracks may extend from front to back of the structure and give rise to leakage.

If the face is curved, even though the arch action is small and cannot be counted on to reduce the cross section, it will tend to prevent temperature cracks forming.

For all but small dams it is advisable to provide proper expansion joints at suitable intervals.

For a maximum temperature of $t_1°$ F. (reservoir empty) and a minimum temperature of $t_2°$, the spacing of the expansion joints should not be more than $\frac{4000}{t_1 - t_2}$ ft., with a maximum of 100 ft. For thin dams this spacing should be reduced to $\frac{3000}{t_1 - t_2}$ ft.

Expansion Joints.—Such joints are shown in fig. 48. In type (1) grooves are cast in one end of the section and coated with bituminous paint or pitch. On casting the next section, tongues are formed filling the grooves. In (2) a steel plate is inserted across the joint. One half of this is painted to prevent adhesion of the concrete. In (3) the tongue and groove of (1) is used with the addition of a bent copper or lead strip as shown. In (4) a

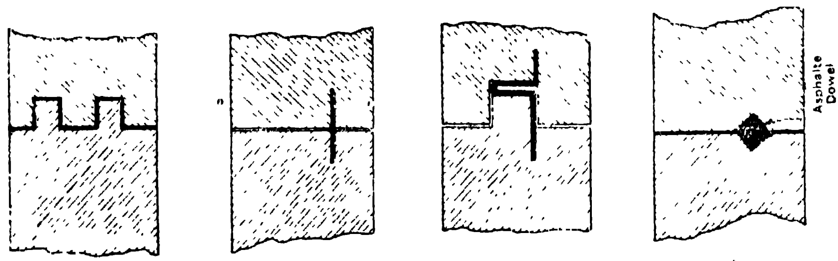


Fig. 48.—Expansion Joints in Masonry or Concrete

square or circular groove is moulded from top to bottom of the joint, and this is filled by a dowel of asphalt, which may be from 3 to 6 in. in diameter. This forms a simple joint and has given good results in practice.*

Materials.—The material to be adopted depends largely on the availability of materials in proximity to the site. Where suitable stone is available and cheap the construction may be of ashlar masonry or of rubble masonry with coursed facings, care being taken to break all joints so as not to produce planes readily susceptible to shear. Where sand and gravel are available, a concrete dam is often the cheapest type. The aggregate may consist of broken stone, but where large blocks are available these are incorporated in the mass. In this case care should be taken that each block is separated from its neighbours by an adequate thickness of cement binding.

58. Buttressed Dams.—These are always of ferro-concrete. The stability of the dam is now due mainly to the vertical component of the water pressure on the inclined up-stream face or deck. This is usually inclined at between 40° and 50° to the horizontal.

As so much of the material is put into an ordinary gravity dam merely to provide weight, it is obviously economical to replace this by water. Further, the distribution of pressure over the foundations can be made

* *Proc. Inst. C. E.*, 1911-2, Pt. I, p. 68

much more nearly uniform, and the possibilities of failure, owing to sliding or overturning, are greatly reduced. These dams can be used in many

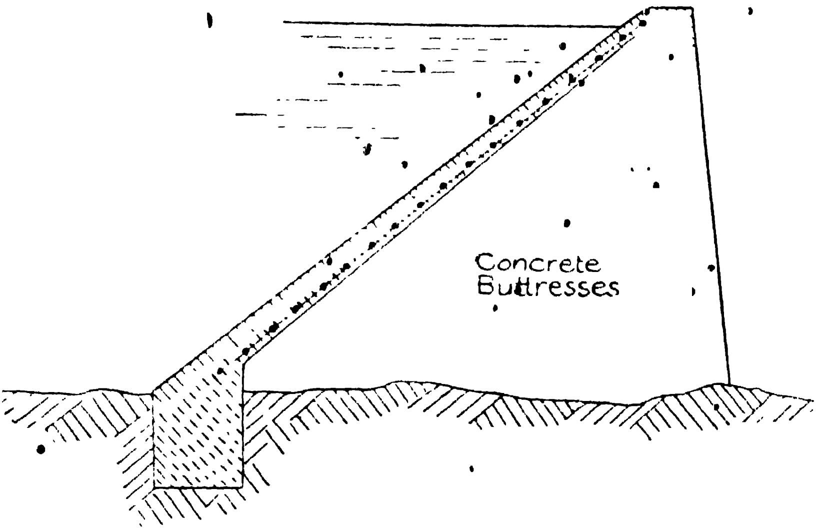


Fig. 49.—Ferro-concrete Dam Section

situations where a gravity dam would be impossible owing to poor foundations.

Figs. 49 and 50 * show diagrammatic sections of this type; the latter for

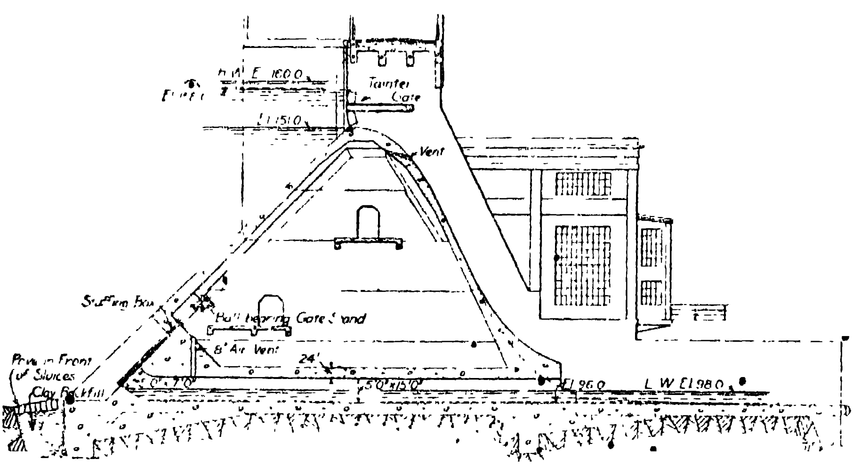


Fig. 50.—Section of Ferro-concrete Dam, showing Spillway.

cases where the dam is required to serve as a spillway, in which case a down-stream deck is necessary.

* By courtesy of the Ambursen Construction Co. This shows the Rapidan Dam, U.S.A. A Tainter gate and a drainage sluiceway are shown in the diagram.

The thrust is conveyed to the foundation by buttress walls, the spacing of which must be settled for each case. The decking requires to be stronger as the pitch of the buttresses is increased, but it must be remembered that almost the only cost involved in thickening up the deck is for the concrete—the forms and reinforcement remain the same—while more buttresses mean more difficult work at low levels, which can only be carried out at favourable seasons.

The thickness of the decking can be reduced by longitudinal girders of reinforced concrete, or by additional buttress walls at right angles to the up-stream deck.

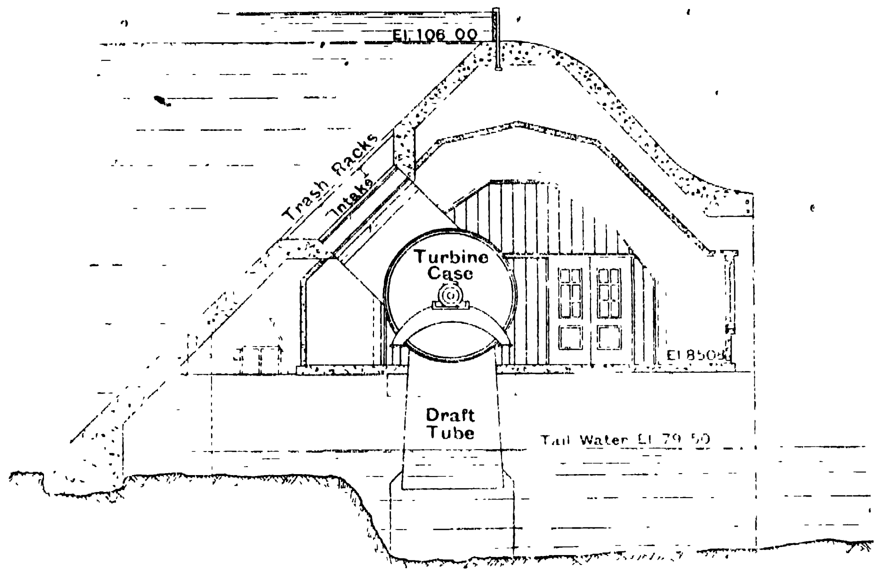


Fig. 51.—Hollow Ferro-concrete Dam forming P

The hollow space under the decking has been used for the power house in a few American and Canadian plants.

Fig. 51 shows a section of such a dam.* The height is 30 ft., and the length 200 ft. It is designed to contain three 500-h.p. units. The power house is enclosed with 4-in. walls of cement plaster on ferro-inclave, entirely separate from the structure of the dam itself. This ensures against any moisture from sweating or condensation. When in flood, the back-water rises almost to the lip of the dam. The breast of the power house is therefore heavily reinforced to stand external pressure. Light is admitted through two framed port-holes in each bay. Entrance to the power house is normally obtained through a heavy steel watertight door at each end of the dam. For entrance during high floods an auxiliary opening is placed in the end buttresses immediately under the crest of the dam, and above the highest water level.

* At Ilchester, Md. By courtesy of the Ambursen Hydraulic Construction Co.

59. Arch Dams.*—For narrow rock gorges, the arch dam offers some advantages. Here the dam is circular in plan with the convex face up stream, and the thrust due to water pressure is transferred directly to the sides of the valley. It is usual to neglect the effect of the weight of the dam in producing stability, in which case, if the section is considered as a portion of a thick cylinder, the necessary thickness at any depth is given by

$$R_1 = R_0\sqrt{\frac{S-P}{S+P}}.$$

Here R_1 and R_0 are the outer and inner radii in feet; S the working stress in tons per square foot; and P the water pressure in tons per square foot. On the assumption that the dam acts as a simple cylinder, the thickness would be given by

$$T = \frac{RP}{S} \text{ ft.},$$

where R is the mean radius, and this simple but inaccurate formula is often used for approximate calculations. Actually the weight of the dam and its attachment to the foundations increase its stability and reduce the stress appreciably, and the working stresses to be adopted when the above formulæ are used are much higher than in a gravity dam. The permissible values are approximately as follow:

Ashlar masonry (granite) ..	..	45 tons per square foot.
,, ,, (hard sandstone)	..	28 ,, ,,
,, ,, (soft sandstone) ..	..	18 ,, ,,
Concrete (depending on aggregate)	..	15–25 ,, ,,

Such a dam is shown in fig. 52.† This has a radius of 150 ft.; base thickness, 108 ft.; top thickness, 10 ft.; height above foundations, 328 ft.; above bed of stream, 243 ft.; distance between abutments, 180 ft.; length of crest, 300 ft. The section is trapezoidal, the front face having a slope of 1 in 4, and the back face 1 in 6·5.

Arched dams are only suitable for comparatively short spans. The radius giving a versed sine of about one-third its length, or an included angle of about 130°, is the most economical. The cross section of such a dam of 500 ft. radius calculated for a stress of 20 tons per square foot is approximately the same as that of a gravity dam for the same head. For moderate spans careful investigation is necessary to determine whether an arch dam or a straight gravity dam is most economical.

Multiple Arch Dams.—These are buttressed dams, with the decking arched between the buttresses. In practice the face is usually inclined, so that the water pressure assists in securing stability. They require less material than the straight deck type, but the form work is increased, and

* See *Proc. Inst. C. E.*, Vol. 178, 1908–9, p. 1. Also B. A. Smith, *Proc. Am. Soc. C. E.* March, 1920. † The Shoshone Valley Project, Wyoming.

it is only in special situations that they will prove less costly. They are not well adapted for passing large volumes of water.

60. Ice Pressure on Dams.—Sheet ice, after its formation, is subject to contraction and expansion due to changes in temperature. The contraction results in cracks which fill with water, which freezes and renews the continuous sheet. A rise in temperature causes the whole mass to

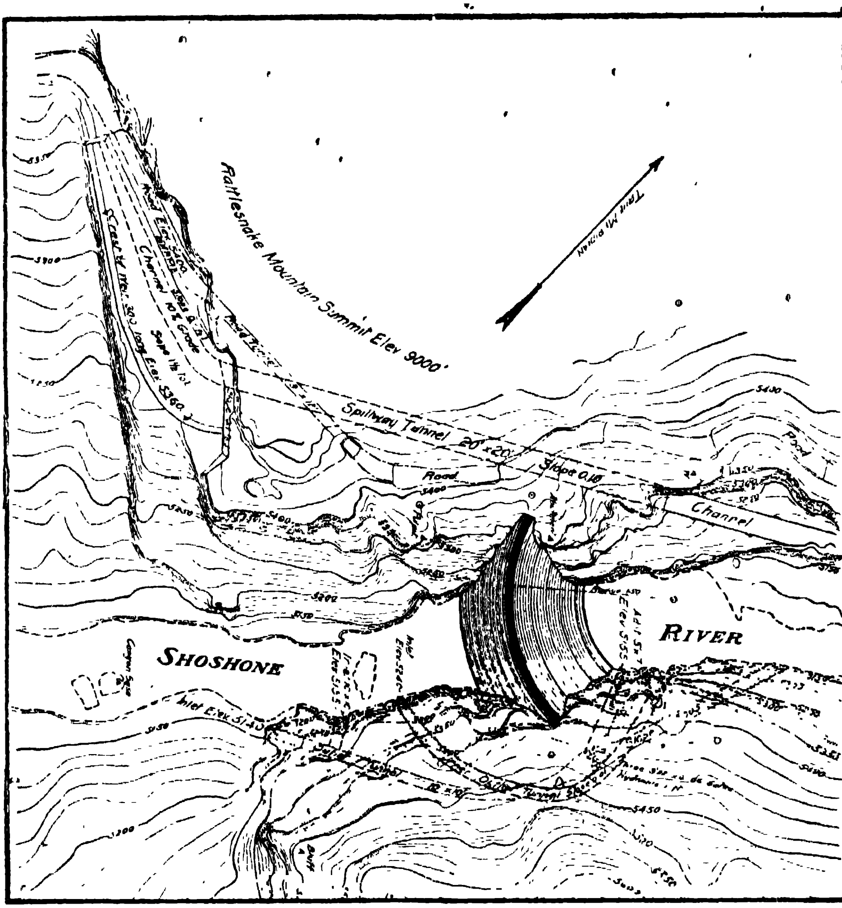


Fig. 52.—Plan View of Shoshone Dam

expand and to exert a thrust wherever its boundaries resist expansion. Where the boundaries are sloping, as in the case of the usual shore line, or of the inclined face of a buttressed dam, the ice sheet slides over the surface and little pressure is produced. Where the boundary is vertical the thrust may be large.

The greatest ice pressures are produced in comparatively narrow waters where the ice is confined between a vertical structure on one side and a vertical shore line on the other. A heavy sheet of ice freezing solidly

to the face of a dam, followed by a lowering of the water level, may also give rise to comparatively large forces on the dam, and any subsequent rise in the water level is apt to produce heavy pressures on the face of the dam.

The magnitude of the ice-thrust, to be allowed for in such cases, is very indefinite.* The thrusts which have been allowed in the design of many recent dams in the United States range from 24,000 lb. to 47,000 lb. per foot run of the highest ice line. All these dams were of the non-overflow type, and the variation in the allowance was due to variations in local conditions and in the type of location. An overflow dam, so long as it is subject to overflow, does not experience any ice-thrust.

61. Dam Accessories.—It is often desirable to be able to vary the height of the crest so as to maintain a fairly constant water level, whatever the volume of water passing over the spillway. This reduces the amount of land covered at times of flood, while at the same time the full head and storage is obtained at times of low flow.

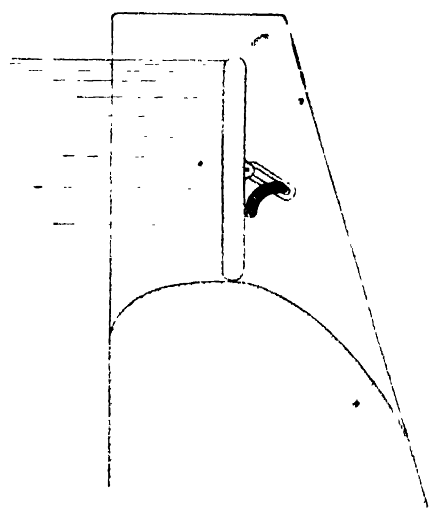


Fig. 53.—Movable-crest Dam

62. Flashboards.—These consist of a row of wooden boards or panels mounted on the crest. They may be held in place by permanent steel supports embedded in the concrete. This involves the removal of the boards before the actual arrival of a flood. Alternatively the supports may consist of iron rods designed to bend and release the boards when the height of the water reaches a certain limit. This is safer but expensive, as the boards must be replaced each season. Flashboards are now seldom used except in minor installations.

63. Movable Crests.—One simple type of movable crest consists of a gate pivoted at a point about one-third of its height above the crest of the dam, as shown diagrammatically in fig. 53. The water pressure maintains the gate in equilibrium until the level overtops the gate, when the gate overturns. By curving the surface of the pivot, so that the point of support rises as the gate tilts, and by weighting the gate itself, the level can be maintained constant within fairly narrow limits. These gates have been used chiefly in America.

In Switzerland a variety of other types have been developed by Barrages Automatiques of Zürich, some of which are shown in figs. 54 to 56. Here the gate is counterbalanced by a reinforced-concrete weight so as to balance the water pressure when the surface is level with the top

* *Trans. Am. Soc. C. E.*, Vol. LXXV, 1912.

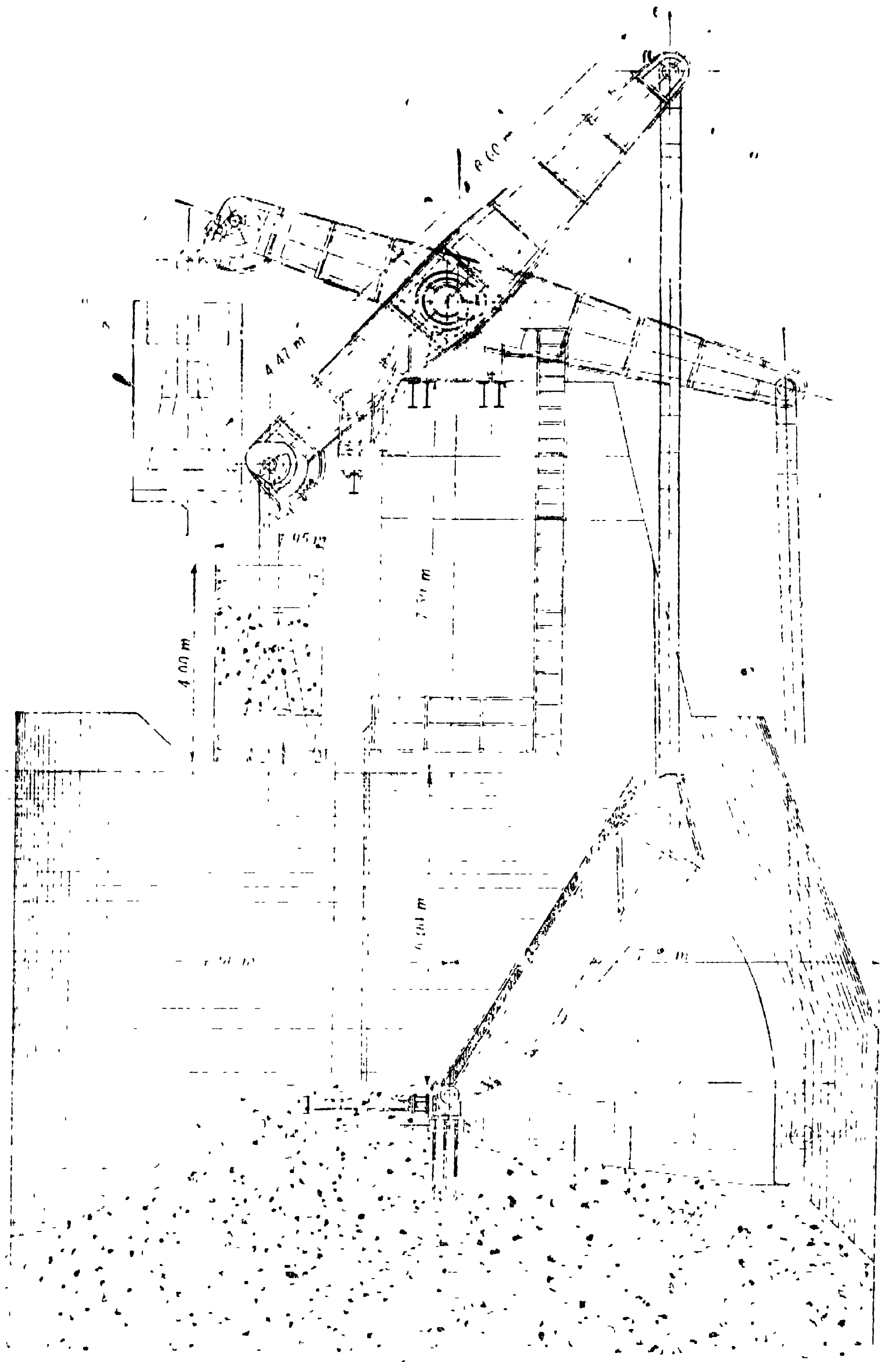


Fig 54.—Automatic Movable Crest Weir
(Side Elevation)

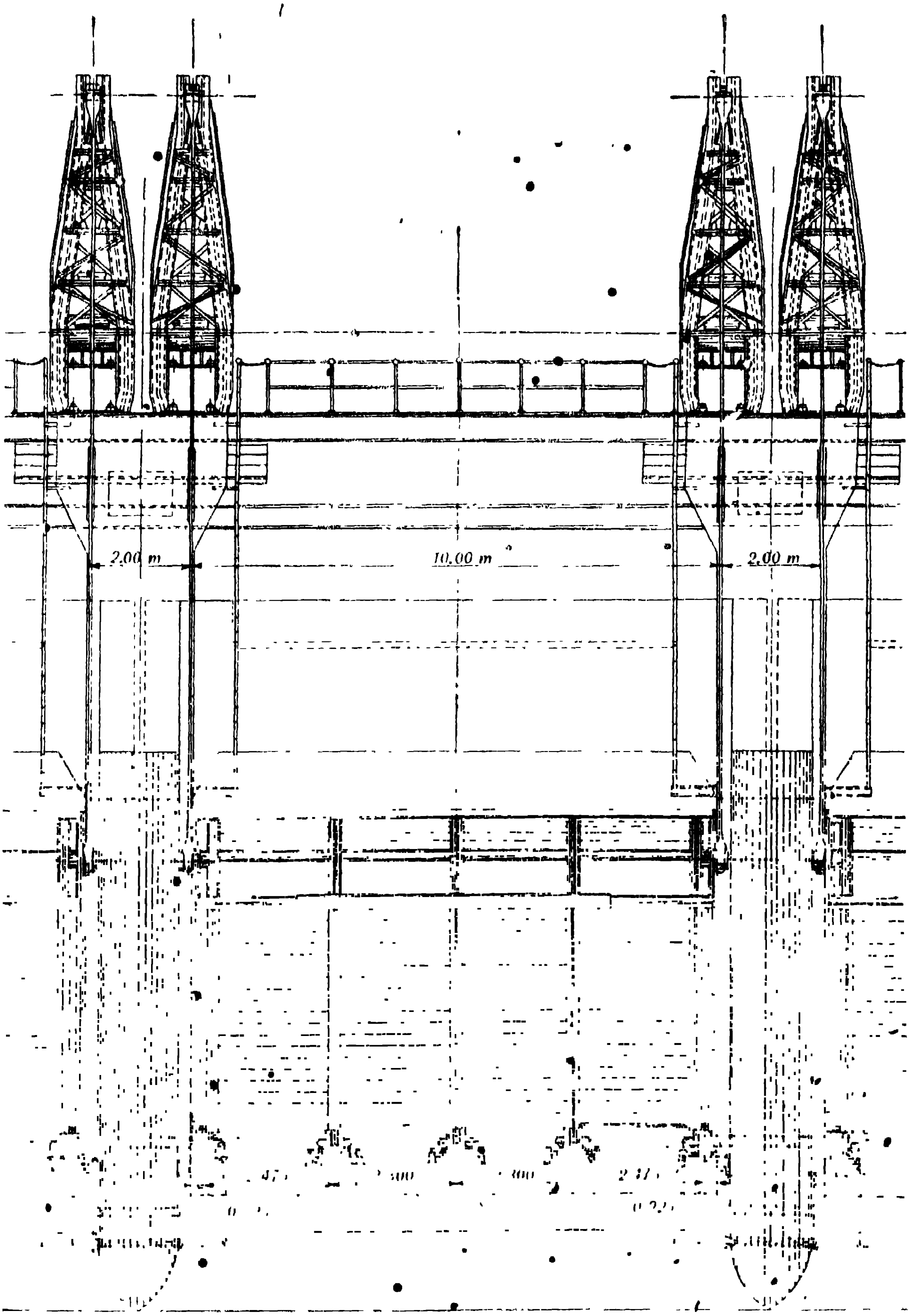


Fig. 54.—(a)
(Front Elevation)

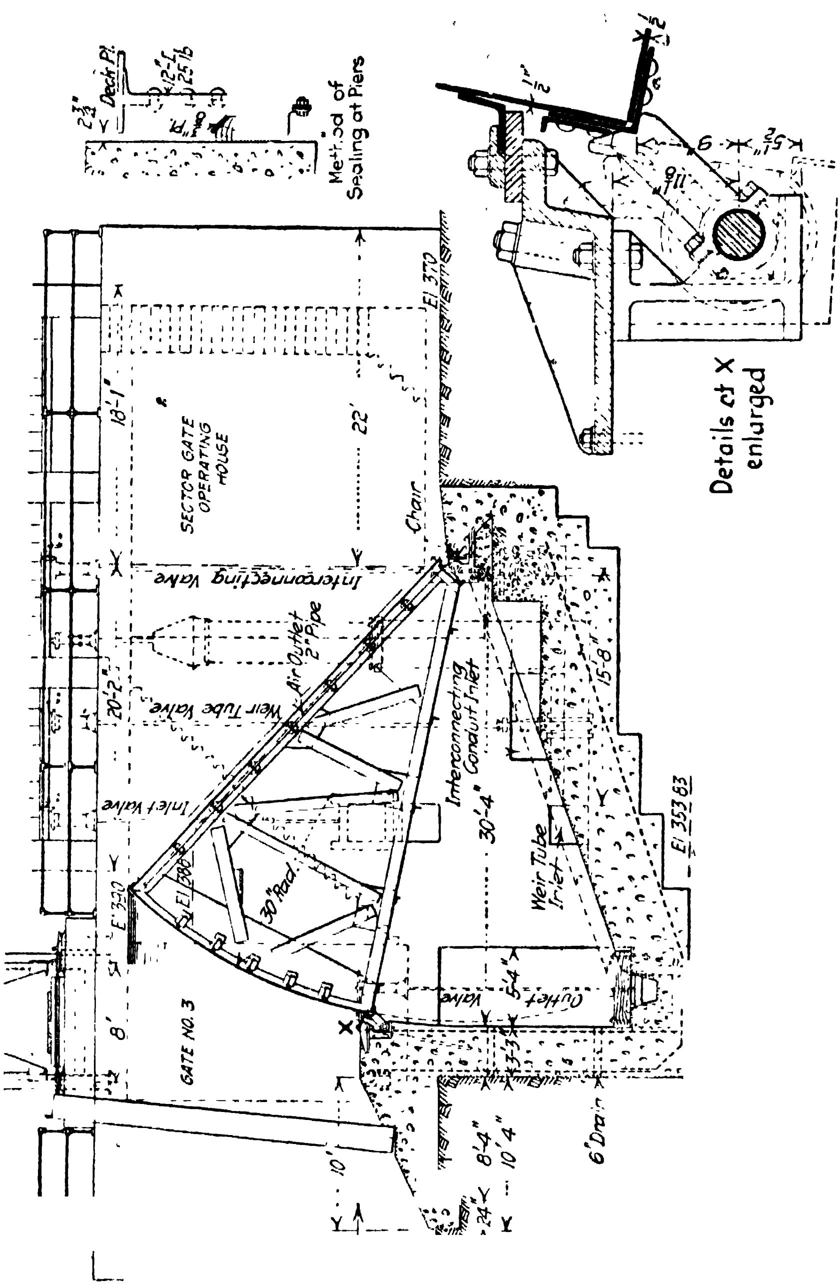
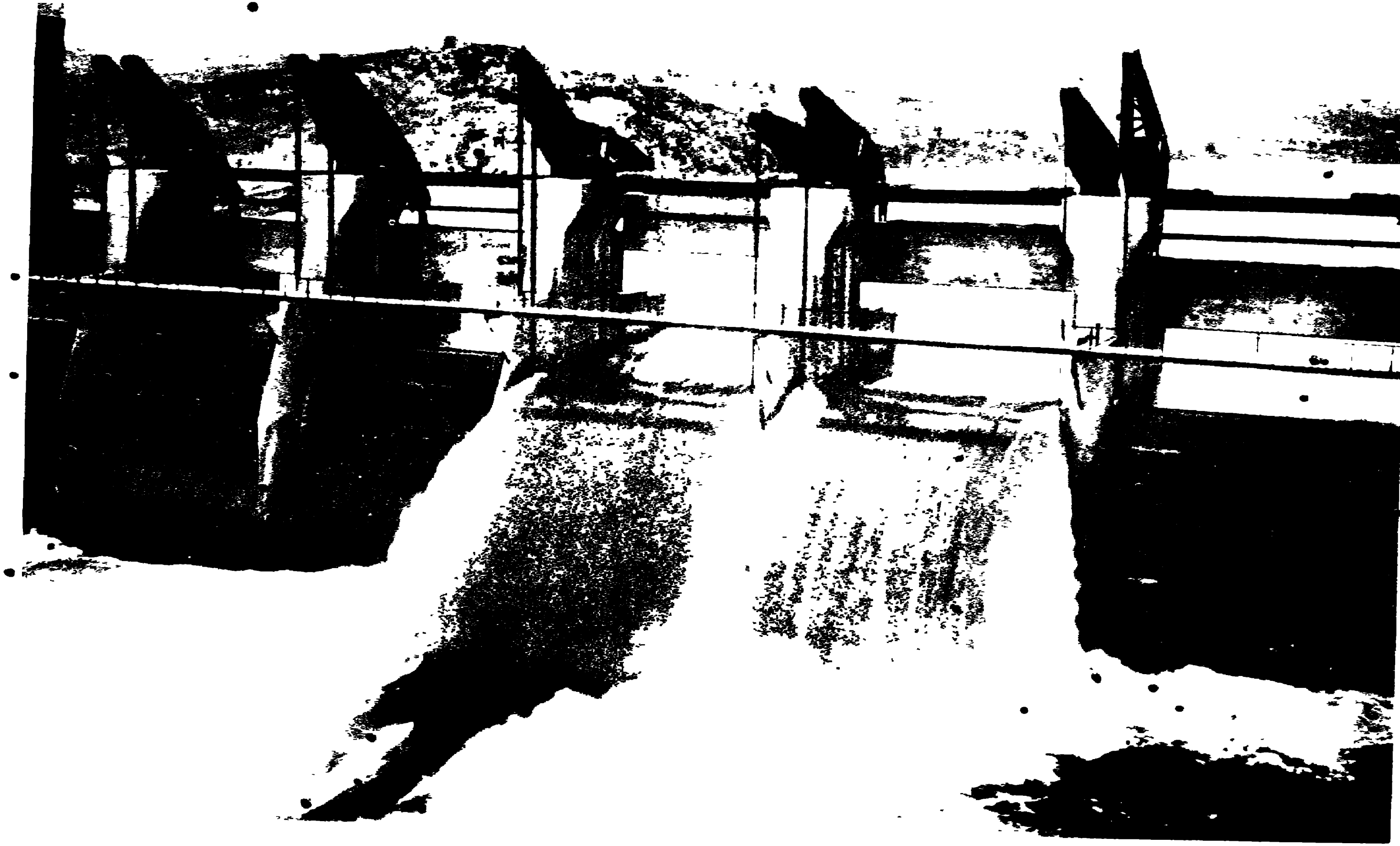


Fig 57 —Sector Gate

of the gate. The extra pressure accompanying any further rise in level depresses the gate and allows the water to spill. These gates enable the level to be maintained within very narrow limits.



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Figs. 54 and 55 show the type most suitable for comparatively low weirs. In this design the piers and counterweights afford some obstruction to the passage of floating debris. Fig. 56 † shows a type suitable for a higher fall. Here there is no such obstruction, but the moving parts are not so accessible. This design is adapted for use on a buttressed dam. Instead of a counterweight hung from levers a rolling counterweight is often used. As the gate is depressed the counterweight is rolled up an inclined surface, and, by suitably varying the inclination of the latter, the water pressure can be very exactly balanced at all positions with a fixed up-stream level. This method is more complicated than that previously described, and the slight advantage in the accuracy of level hardly justifies the extra cost.

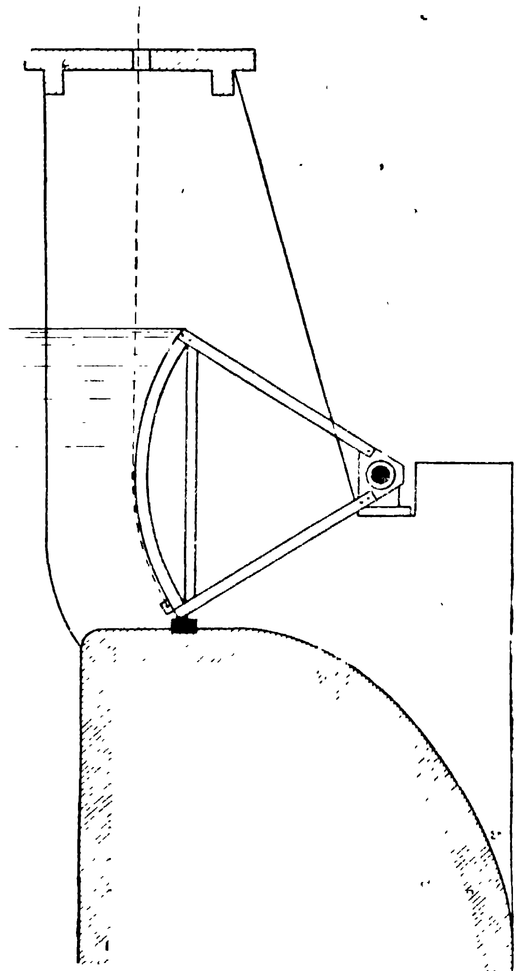


Fig. 58.—Tainter Gate

Sector Gates.—Fig. 57 * shows a sector gate as fitted on the Genesee River at Rochester, N.Y. Each gate consists of a sector of a cylinder, 60 ft. in diameter and 100 ft. long. One radius, and the curved portion of the sector, is covered by steel plating. The structure is pivoted at its centre, and is raised or lowered by varying the height of the water in the chamber underneath the dam. When the gate is in its lowest position and the water is level with the crest of the dam, the upward pressure is just sufficient to relieve the weight of the sector. As the gate rises, its centre of gravity moves through an arc, so that its horizontal distance from the pivot decreases. Any increase in water level, if accompanied by a corresponding increase in the pressure below the gate, will therefore lift the gate, and its height is regulated by regulating the latter pressure by means of a special valve.

Tainter Gates.—These gates are very largely used owing to their ease

* *Engineering News*, 8th March, 1917, p. 39? † *In pocket at end*

of operation. As indicated in fig. 58, each gate consists of a pivoted steel framed sector, which is lifted by a chain or cable from the track above, either by a portable hoist or by a permanent winch. To ensure water-tightness, the bottom edge of the gate bears against a timber sill, let into the crest of the dam, or alternatively the bottom edge of the gate carries

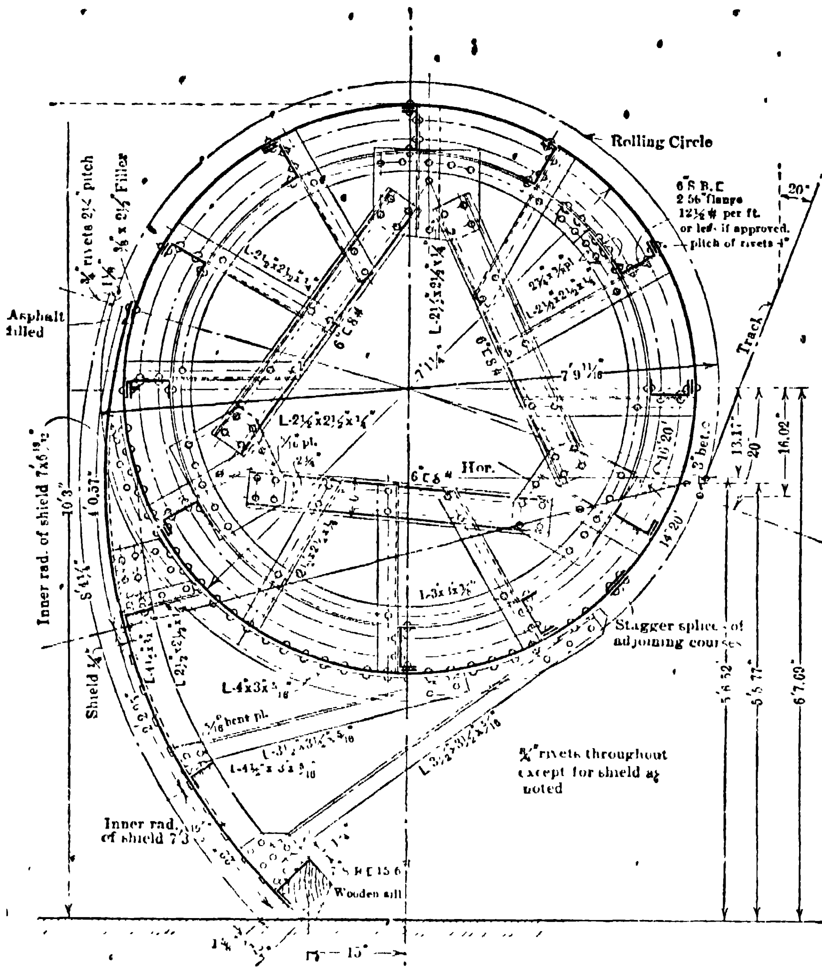


Fig. 59 Rolling Dam on Grand River Colorado

a timber sill, which bears against the edge of a steel plate let into the crest of the dam. The chief drawback to this type of gate is that, to support the pivot, the piers must be extended a considerable distance down stream from the crest.

64. Rolling Dams. A type of construction which has been used in Germany, and, to a limited extent, in the United States, consists of a large hollow cylinder of boiler plate. This rests normally on the

crest of the dam, and is raised when required by rolling it up an inclined plane. For this purpose each end of the roller may be provided with a gear engaging a rack laid in a recess in the pier. A sprocket chain or cable encircling each end of the roller and connected to the operating mechanism enables it to be raised as required. When built in large sizes the roller is usually made of a smaller diameter than the height of the weir. The up-stream face of the roller is then provided with an extension as shown in fig. 59, which shows the roller for a 70-ft. dam of this type on the Grand River, Colorado.

The advantages of this type of dam are the great length (up to 150 ft.) possible for each section, and the small number of piers which are necessary; the ease and speed of operation; and the tightness of its closure.

65. Sliding Gates.—Sliding gates are not so generally used in connection with the dam itself as for the intake, and will be described in a later section. The one exception is the Stoney Roller Gate (fig. 60).* Here the water pressure is transmitted to the piers through a series of rollers. The friction is very small, and such gates are therefore suitable for large openings and for operation under high heads. They are specially suitable for drainage gates. Fig. 60*a* shows in some detail the arrangements for preventing leakage between the gate and the piers.

All such sliding gates are open to the objection that the hoisting gear must be placed high enough to enable the gate to be lifted clear of the top of the water, so that the height of the hoisting bridge above the crest of the dam must be at least twice that of the gate.

66. Drainage Gates. The dam must be provided with means for drawing off the stored water for inspection and repairs. Gates must therefore be provided at the lowest level, and these should be sufficiently capacious to discharge appreciably more than ordinary low-water flow of the river; otherwise too much time will be wasted waiting for the dam to empty, or it may be impossible to maintain the level sufficiently low except under very favourable conditions. These gates are also of great use during the construction period, as they are available for passing the flow while the lower portions of the dam are being erected. Such gates must operate under high pressures and must be reliable. The Stoney gates already referred to are especially suitable. If the flow to be dealt with is small, sluice valves of the ordinary types can be used.

In addition to the accessories required for the proper protection and control of the dam, others may be required for the preservation of the rights of other users of the water. Locks may have to be provided for navigation, while fish-ladders and log ways are required in some countries. The requirements of each locality in these respects must be studied, and the designs modified to comply with any necessary regulations.

67. Intakes.—The intake comprises all the works necessary for drawing the water from the storage basin or river, and delivering it to

* Yadkin River Development, *Engineering News*, 16th Nov, 1916, p. 921

the canal, pipe lines, or other water conductors. If there is no canal or pipe line the intake also forms the forebay.

The site for the intake must be carefully chosen, and the angle of its mouth, with respect to the direction of the current, must be arranged so as to avoid the accumulation of debris. In general, the opening should be approximately at right angles to the current. If the stream carries much floating material, a log boom is useful. This consists of a chain of logs each end of which is anchored to the shore. The logs are fastened end to end, and the boom deflects floating debris and ice away from the intake to special spillway gates provided on the crest of the dam.

Log runs are necessary in most lumbering countries, to enable logs to be conveyed past the power station and head works. The log run usually consists of a V-shaped flume of timber or concrete. This leads directly from a forebay provided with a sluice gate, into the tail race. The season during which logs are moved is usually that of the spring floods, when surplus water is available for this purpose.

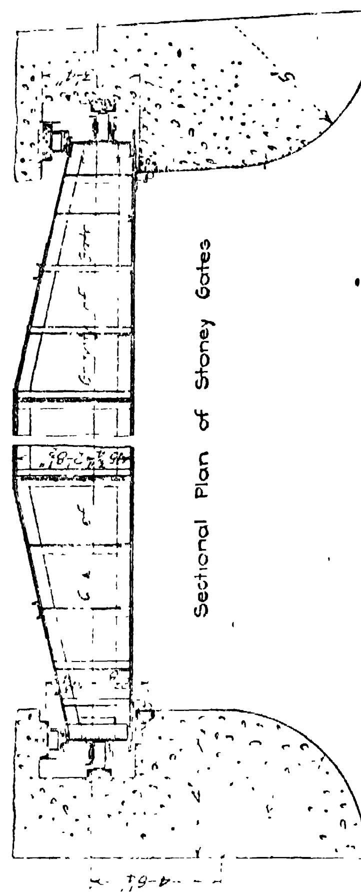


Fig. 60.—Sectional Plan of Stoney Gates

Fig. 61 shows a typical intake, and fig. 62 a typical forebay. As shown in the latter figure, vertical slots are often provided in front of the scour gates or racks for stop-logs. Should the head gates fail to operate, or should it become necessary to repair the gates or the racks, the water can be shut off by sliding the stop-logs down into these slots. The stop-logs consist of heavy baulks of timber, whose ends are sometimes shod with steel to reduce friction and wear. They are made tight against leakage by bags, cinders, straw, &c.

Intake Racks and Screens. The intake must be provided with a coarse rack or screen to catch floating debris. Fig. 63 and 63*a* show a typical screen, consisting of a series of flat iron bars on edge, 2⅝ in. pitch, and 3 in. deep by ⅜ in. thick. The spacing depends on the kind of debris to be expected, but should not exceed 4 in. The bars are made in sections for convenience in placing, and must be firmly supported so as to be capable of withstanding the pressure should the whole face become choked. In order to

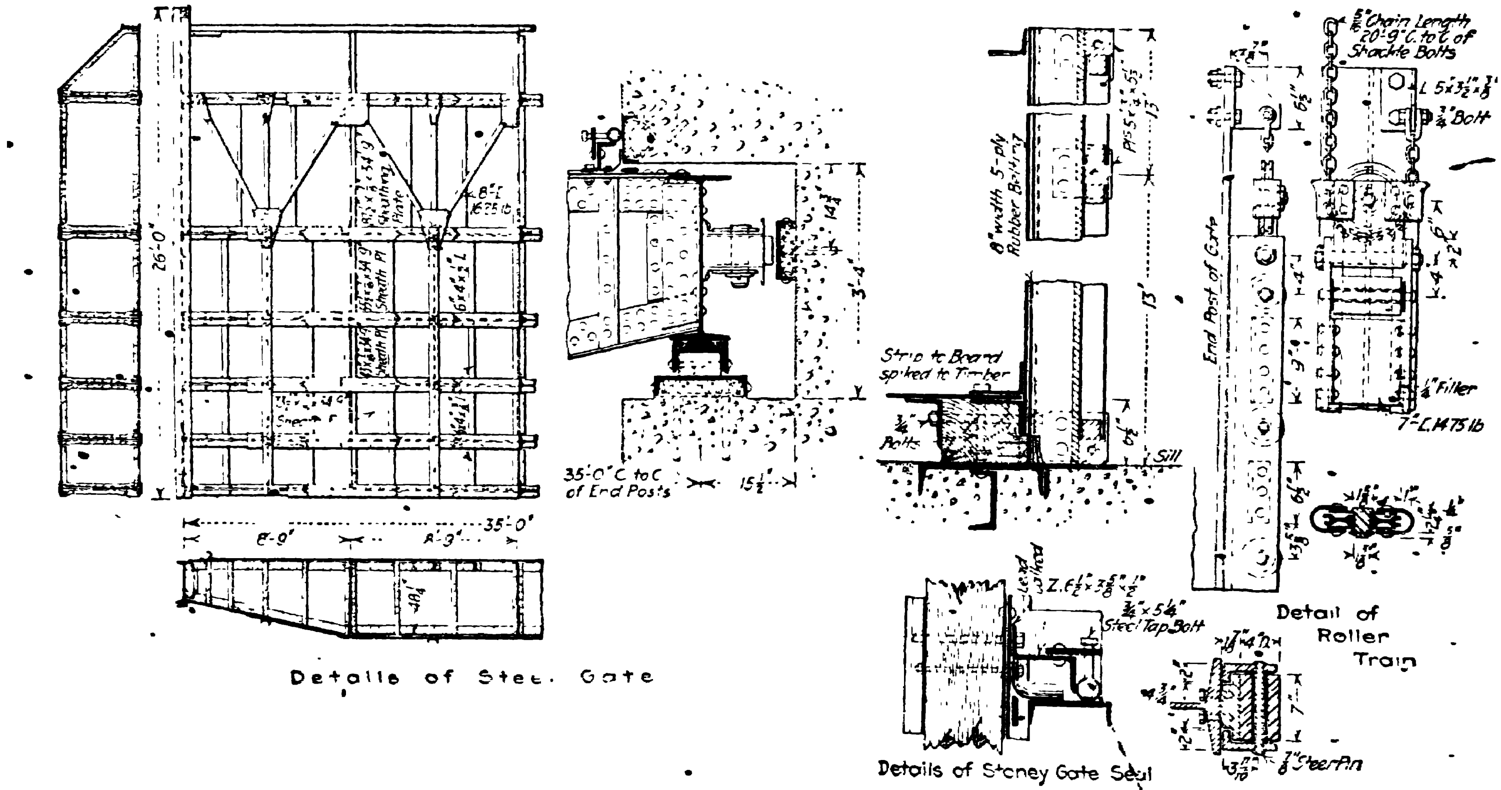


Fig. 60a.—Stoney Sluice Gate and Details

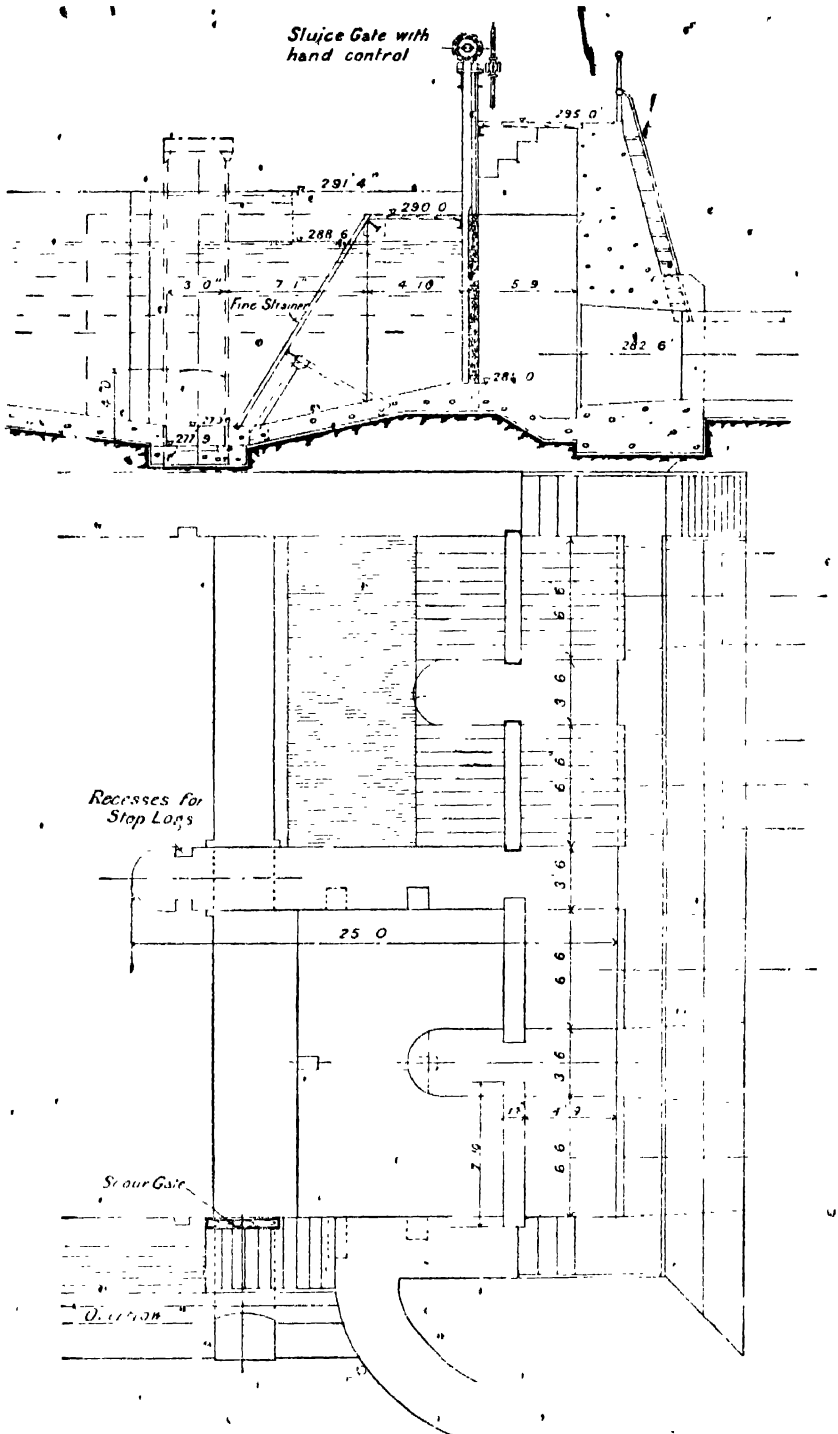


Fig. 62.—Typical Forebay for Water-power Plant

increase the area of the waterway, and to facilitate raking operations, they are usually placed at an angle of about 60° to the horizontal. It is advisable to have separate racks for each opening, so that one may be repaired without affecting the operation of the remainder. The racks in the forebay should be of finer pitch than those at the intake, in order to catch any small debris passing the intake screen, and any leaves, &c., which may enter the head race between the intake and the forebay. The pitch should not exceed about ·1 in.

Where the racks are very deep, or where the amount of debris is large, hand cleaning becomes expensive. Mechanical raking devices may be used in such cases. Strainers of the roller type, in which a fresh surface is continually being presented to the stream, have been used, but the wear and tear on this type is relatively great.

68. Ice Troubles.—In very cold climates ice is found in the form of sheet ice, anchor ice, and frazil. Anchor ice consists of an agglomerated mass of coarse crystals clinging to the bed of the channel, and formed there by cooling of the bottom due to radiation. This can only occur with a clear sky, and where the surface ice, if present, is transparent. Surface ice will in general prevent its formation. It is usually found in rapid streams, where surface ice cannot form.

Frazil consists of fine spicular ice crystals floating in the water, and formed by a slight supercooling below 32° F. This adheres to the surface ice and to the anchor ice, and may finally choke up the stream. In engineering works, frazil ice is particularly objectionable, as it adheres to the racks and strainers and to the gates of a turbine, and if, due to exposure to air, these are slightly supercooled, freezes into one solid mass.

This trouble can, however, be prevented by a slight heating of the racks or turbine gates. At the Ottawa Electric Company's Power House No. 1 a line of steam pipes laid above water-level and against the face of the rack was found to answer perfectly. Electric heating of the racks has also been tried successfully, 600 amperes at ·3 volts quickly removing the ice from a single rack bar with the air temperature at 15° F.* These bars were $\frac{5}{16}$ in. thick and 18 ft. long. In the same installation, consisting of three 39-in. wheels, 30-ft. head, using 100,000 cu. ft. of water per minute, steam was supplied by a small pipe to each of the wheel housings when the unit began to lose capacity. To supply this, 20 tons of coal were used during four months of the winter, with eleven days on which frazil was bad, only occasional injection of steam being found to be necessary.

Tests show that, by discharging compressed air through small pipe orifices near the bottom of the head race, ice troubles can largely be prevented. The air bubbles create a circulation which tends to prevent freezing. Tests on the Keokuk Dam on the Mississippi River showed that 2 c. ft. of free air per minute, discharged at a depth of 18 ft., would open an area of about 20 ft. in diameter in ice from 10 to 22 in. thick in four days.†

* *Ice Formation*, Barnes (Wiley & Son, N.Y., 1907).
† *Engineering*, 20th Sept., 1918, p. 324.

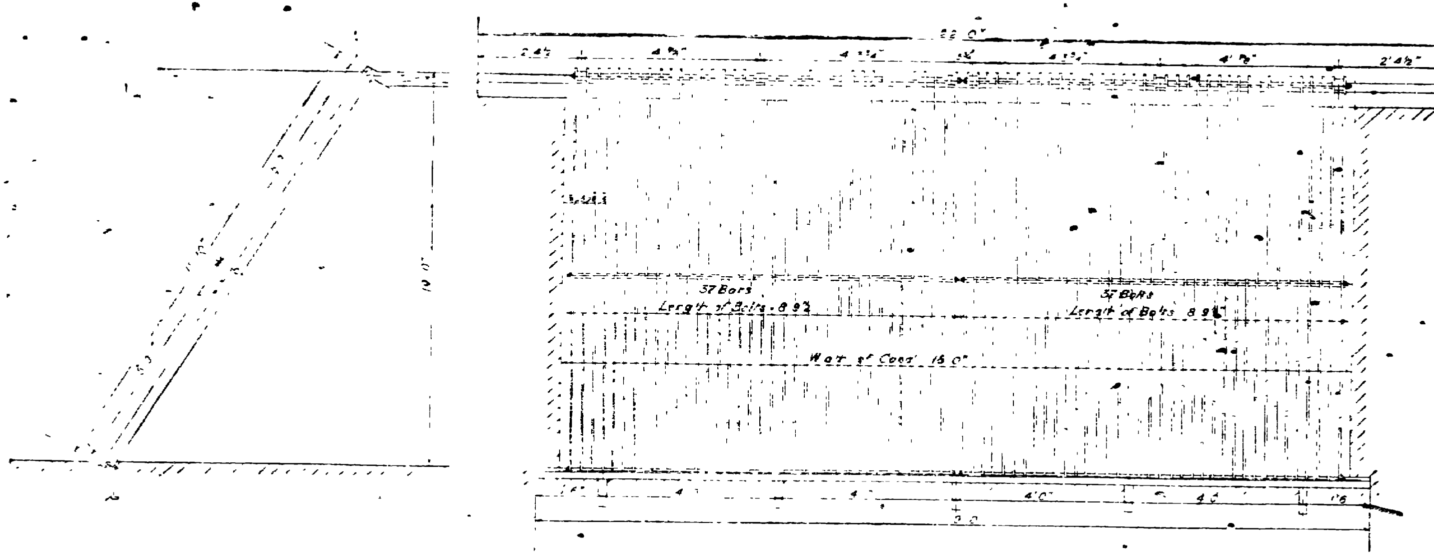
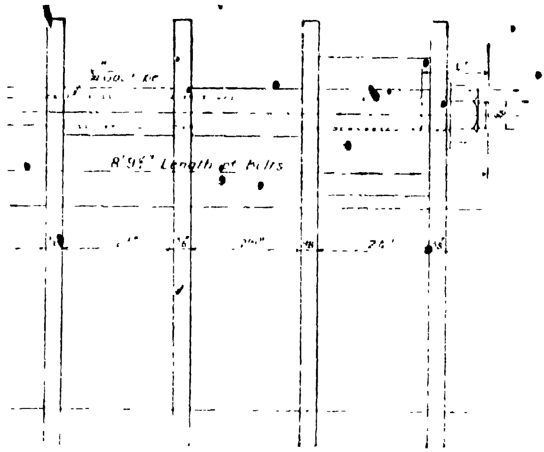


Fig. 63.—Trash Rack for Intake



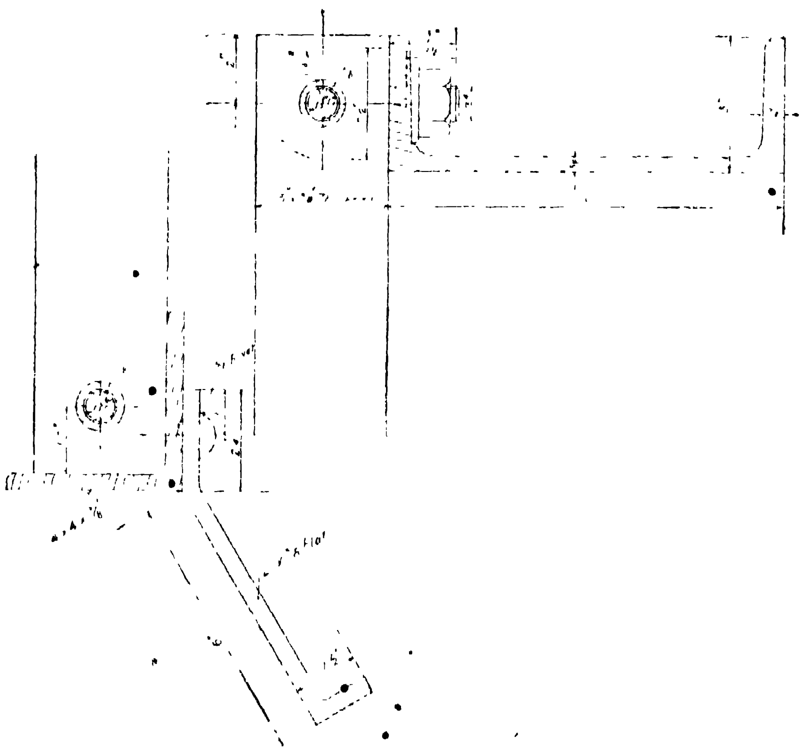


Fig. 63a – Details of Trash Rack

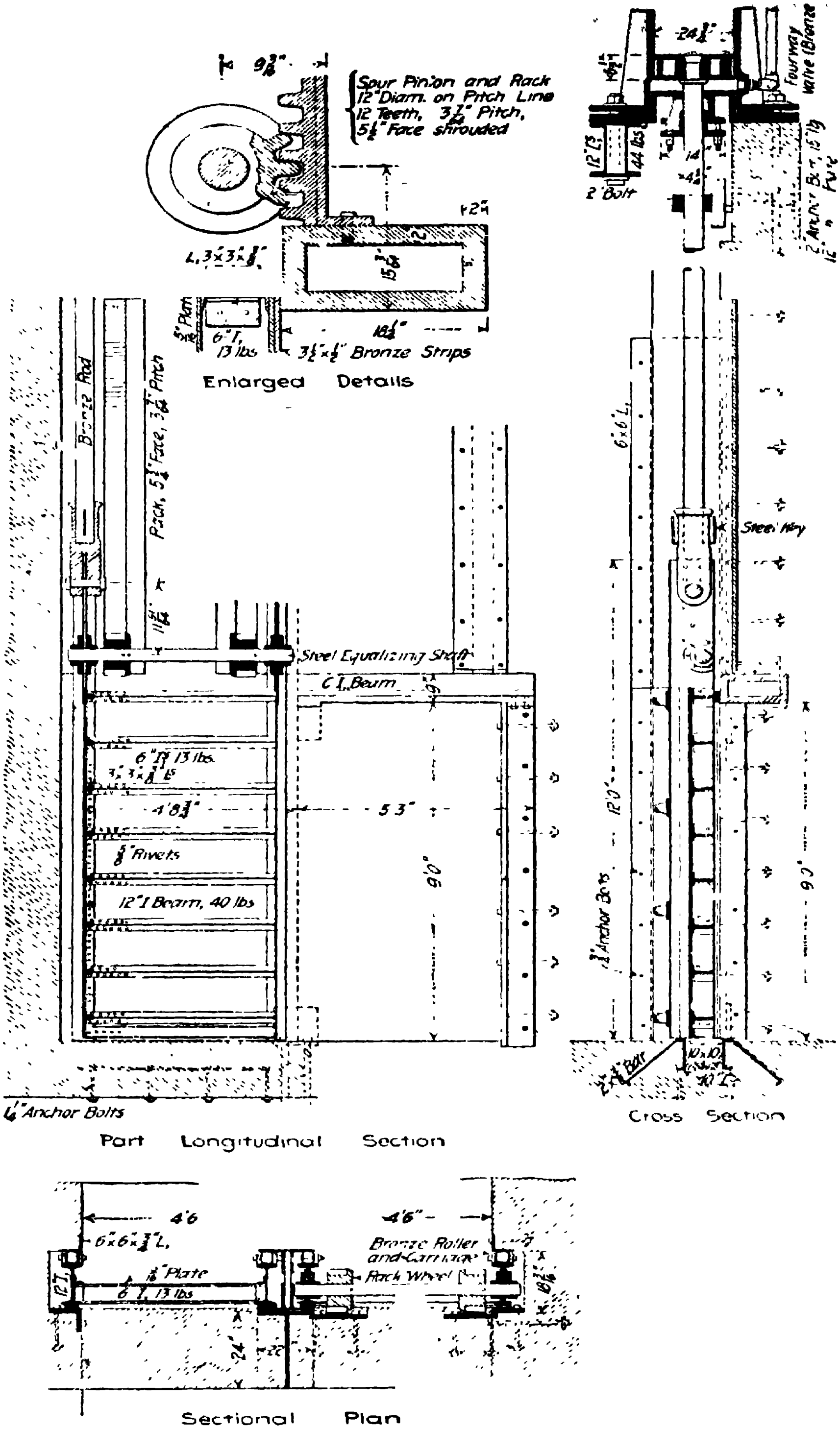


Fig. 65a.—Head Gate operated by Hydraulic Cylinder

In a power plant it is usually advisable that the water at the intake should be covered with surface ice, as being the most effective preventive of the formation of both frazil and anchor ice. If there are large stretches of open water above the surface ice, frazil is formed and adheres

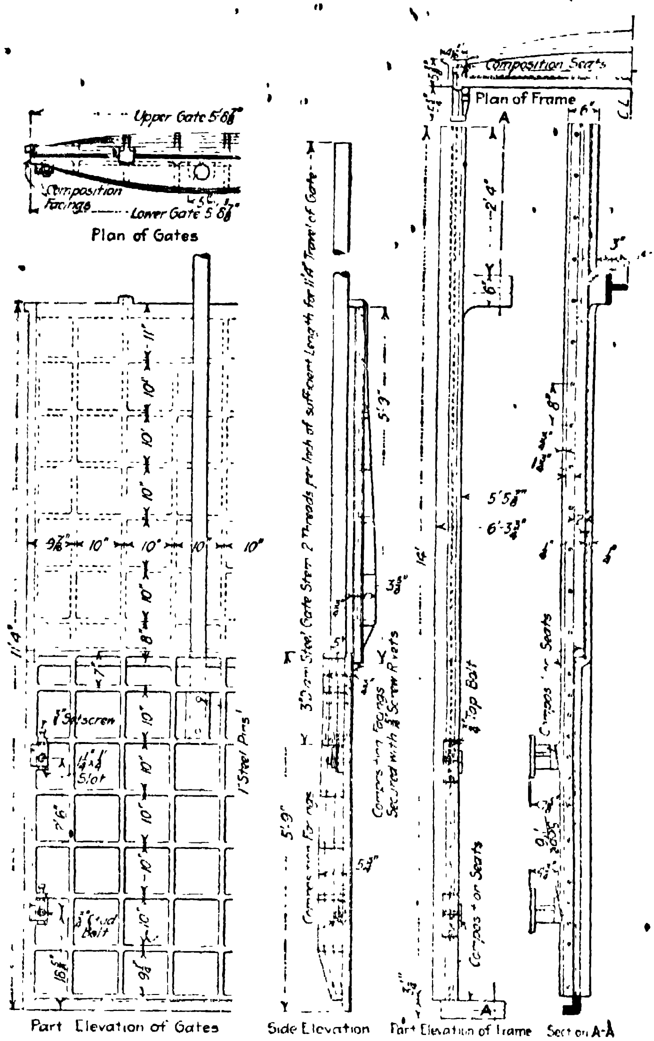


Fig. 66.—Sluice Gate, constructed in two halves

to the lower surface, and may result in a stoppage of the channel. In such a case, or when located at the foot of the rapids, it is better to construct a head race of sufficient size to serve as a settling basin for the ice drawn in. Even then it may sometimes be necessary to blast a channel in the surface sheet. Where a long narrow channel is fed from a stretch of open water, the ice difficulty becomes very great. A surface covering is then harmful, as encouraging the adherence of frazil.

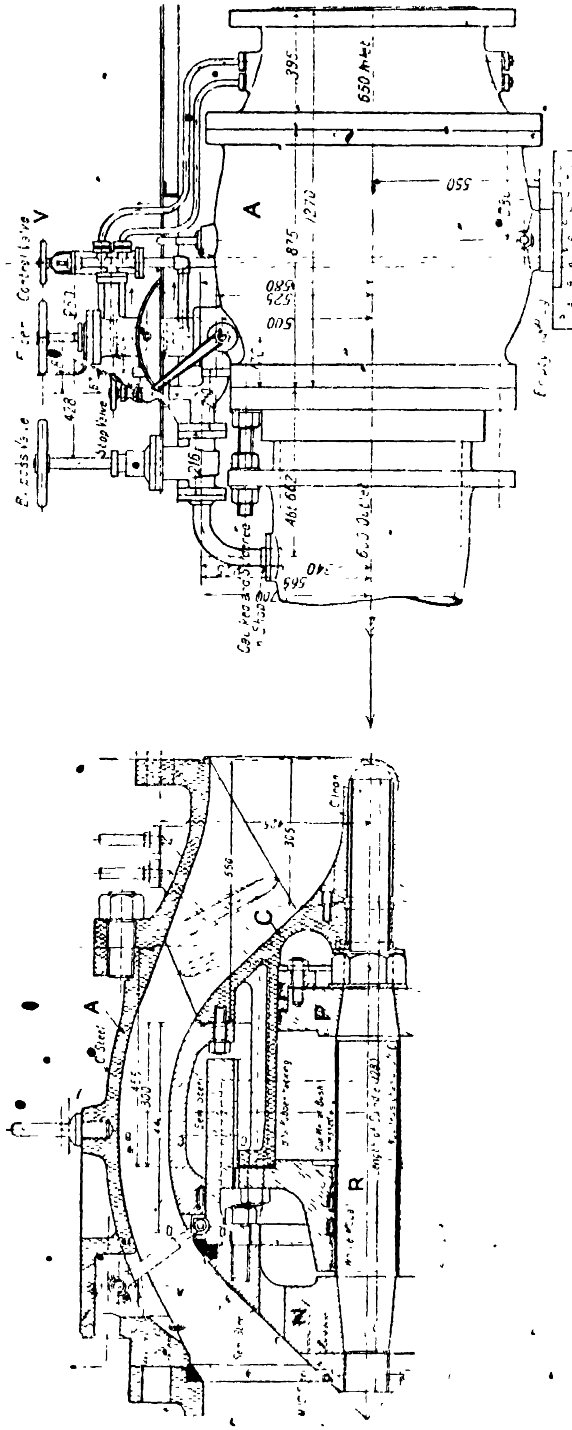


Fig. 65.—Johnson-Boving Power-operated Valve

The break-up of the sheet ice in spring often causes serious trouble in power-plant operation. Where large masses of ice, together with other miscellaneous drift, are carried downstream in flood-time there is always a tendency towards the formation of an ice jam, which banks up the water until the pressure becomes sufficiently great to sweep away the obstruction, while jams below a power station raise the tail-water level and may flood out the plant.

69. Head Gates. Fig. 64* shows a simple type of head gate with hand lifting device. Such gates may be used for sizes up to about 10 ft. by 10 ft. without difficulty, but for larger sizes power-operated gates are desirable. Fig. 65* shows such a motor-operated gate. But electric power may not always be available, and in any case the gates require to be opened before the station is started

* *In pocket at end of volume.* By courtesy of Messrs. Boving & Co., Ltd.

up, so that emergency hand gear is desirable. Fig. 65*a* shows a gate operated by a hydraulic cylinder supplied with pressure water from an auxiliary pump. For dealing with large volumes of water, a number of gates are used side by side. Filler gates as shown in fig. 64 are sometimes possible. As their name indicates, these are by-pass gates formed in the body of the main gate, and are used to equalize the level on the two sides of the main gate before opening the latter. Where only a relatively small amount of water requires to be passed for this purpose, such gates are

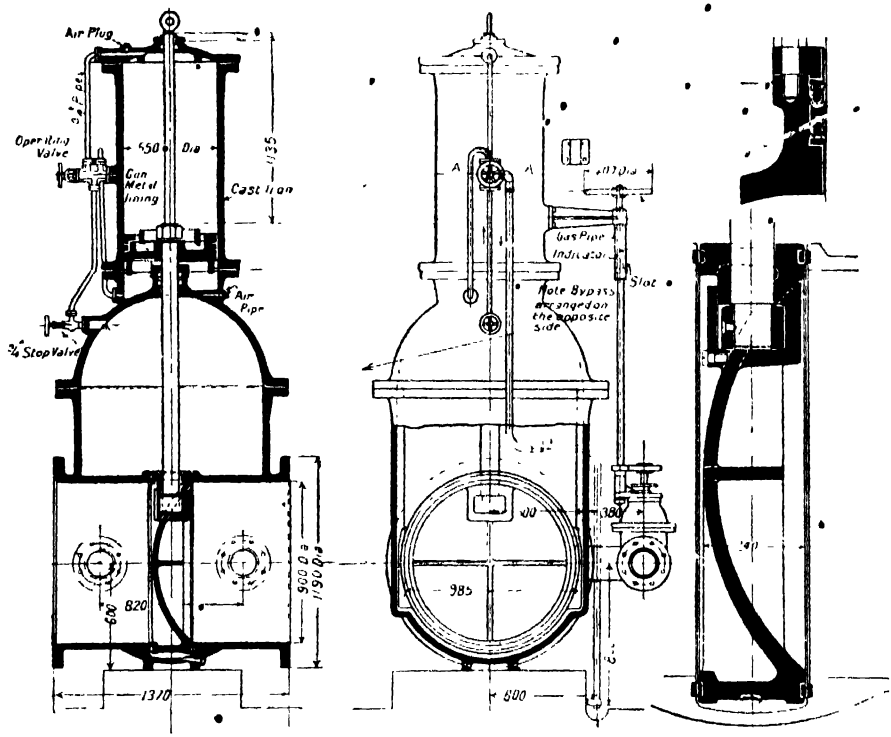


Fig. 68. Sluice Valve operated by Hydraulic Cylinder

very useful, and enable a much lighter main operating mechanism to be used, but the great length of waterway to be filled often prohibits the use of this expedient.

Fig. 66 * shows another type of gate, which is made in two halves. The lower half is lifted first, and by its opening tends to equalize the pressures on the two sides. When it has been raised through its own height, it engages two lugs carried by the upper half of the gate, and both halves are thereafter lifted together.

70. Where the water is led directly into a pipe line, any of the ordinary types of throttle or sluice valves may be used. For large valves of this class operating under heavy pressures, power operation is advantageous.

* *Engineering News*, 16th March, 1916, p. 517.

Fig. 67* shows one type of power-operated valve, of the Johnson-Boving type, as applied to a pipe line 650 mm. in diameter. The valve consists of an outer shell A to which a central casing C is attached by a series of ribs. This casing forms a cylinder fitted with a piston P, whose piston-rod R carries a nose-piece N, which forms the valve. By means of a small control valve V fitted on the top of the main valve, pressure water from the penstock may be admitted to either side of this piston, the other side being at the same time connected to the exhaust. Holes are cored through the body of the nose-piece so that the pressures on this, with the exception of those acting on the piston, are always balanced.

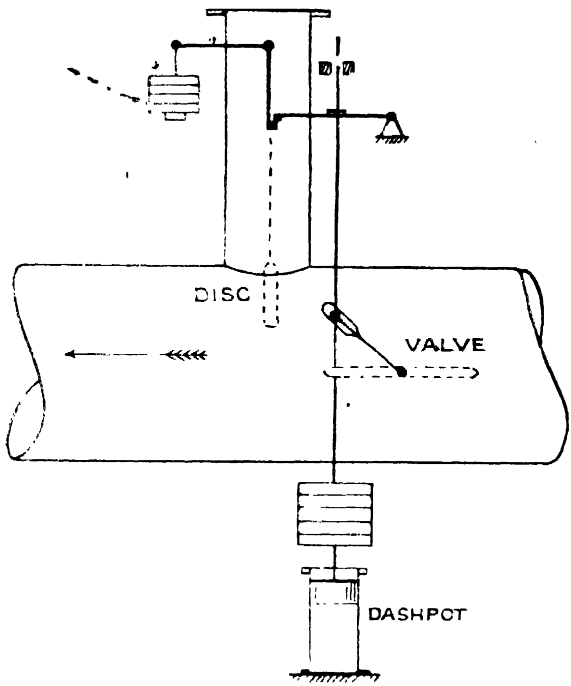


Fig. 69.—Automatic Valve.

Fig. 68 shows another simple type of power-operated sluice valve.

Fig. 69 shows an automatic valve, designed so as to close slowly in case of a rupture at any lower point in the pipe line. Here a flap placed in the water way operates a trip mechanism which controls the main valve disk. Weights on an external drum mounted on the shaft carrying the disk tend to rotate this into the closed position, and should the velocity of flow exceed a definite predetermined value, the pressure on the flap trips the gear and the main valve closes. Its rate of closing is regulated by a dash-pot.

Drainage. A drainage valve should be provided both for the intake and for the forebay. Fig. 70 shows a type of scour gate suitable for this purpose.

71. Channels and Pipe Lines.—The type of waterway used to convey the water from the intake to the turbines depends essentially on the local conditions. Where an open channel is used, this may take the form of a canal or a flume. Where a closed waterway is necessary this may take the form of a tunnel or a pipe line.

Canals. Where the contours are favourable, the excavation not too difficult, and the soil fairly impervious, an open unlined canal is often

* By courtesy of Messrs. Boving & Co., Ltd.

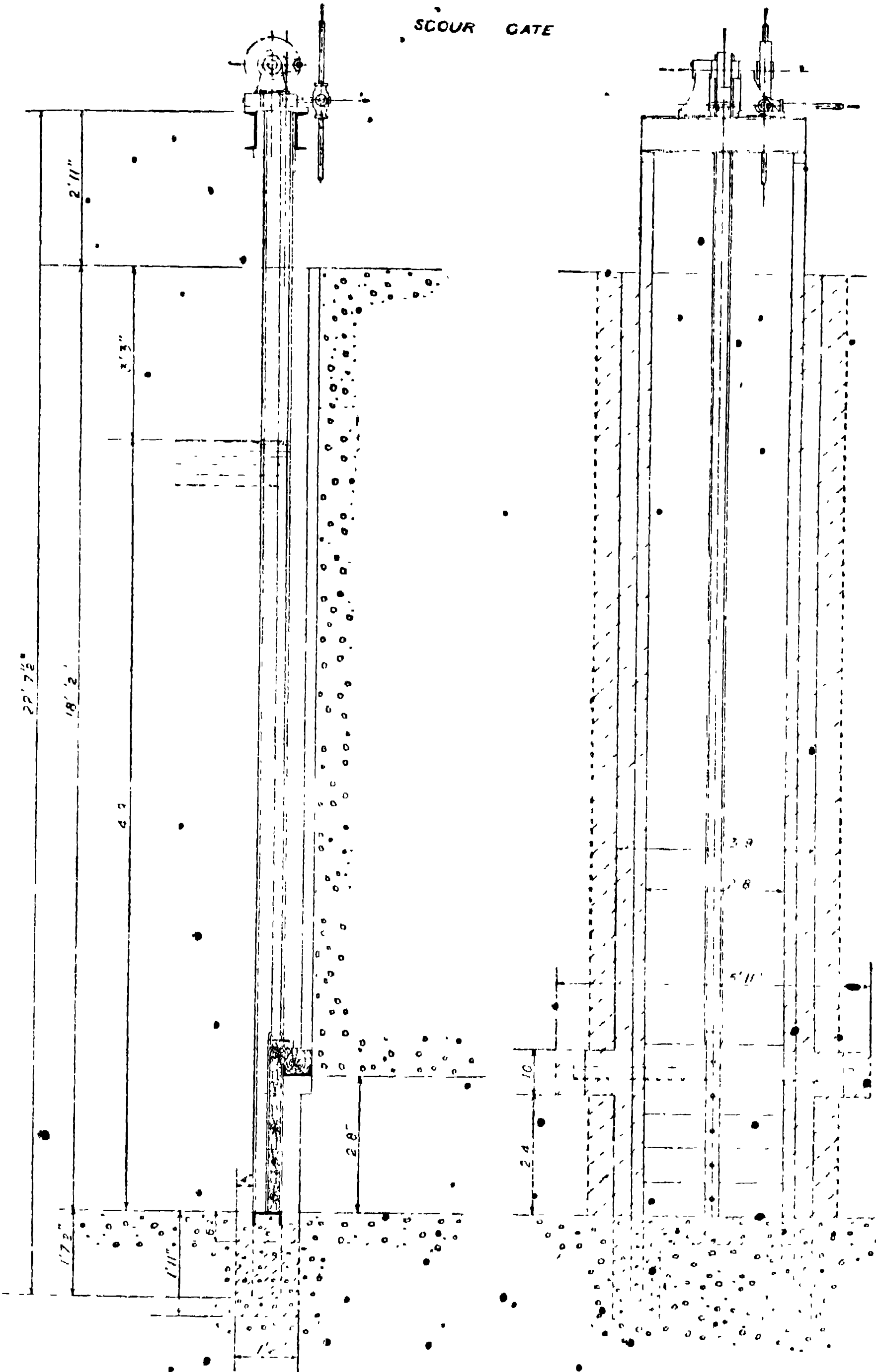


Fig. 70.—Scour Valve

used. In a long canal, especially if the depth is moderately great, seepage requires to be taken into account. Measurements on earthen canals having depths from 4 to 8 ft. show that, except in exceptional cases, where the ground is fissured, the loss varies from 0·6 to 3·6 c. ft. per minute, per 1000 sq. yd. of wetted surface. The average leakage from unlined irrigation canals is about 1·5 ft. of depth per day; in sandy soil from 1 to 2 ft.; in compact alluvial soil about 0·6 ft.; and in gravel from 3 to 5 ft. Where the ground is very pervious the bed and slopes may be puddled, or a concrete lining may be used. This, by reducing friction and the tendency to erosion, enables a smaller section to be used. The concrete lining should be between 3 and 6 in. thick, depending on the size of the canal and on the conditions, and drains must be provided at intervals to prevent the ground water producing dangerous hydrostatic pressures on the back of the lining when the canal is empty. Such a lining has the further great advantage that growth of vegetation is prevented. The cost of a concrete-lined canal of moderate size is approximately three times that of an unlined canal of the same size. On the other hand, with the same friction loss the lined canal need only have one-half the cross-sectional area of the unlined canal, and as its maintenance costs and the cost of cleaning are

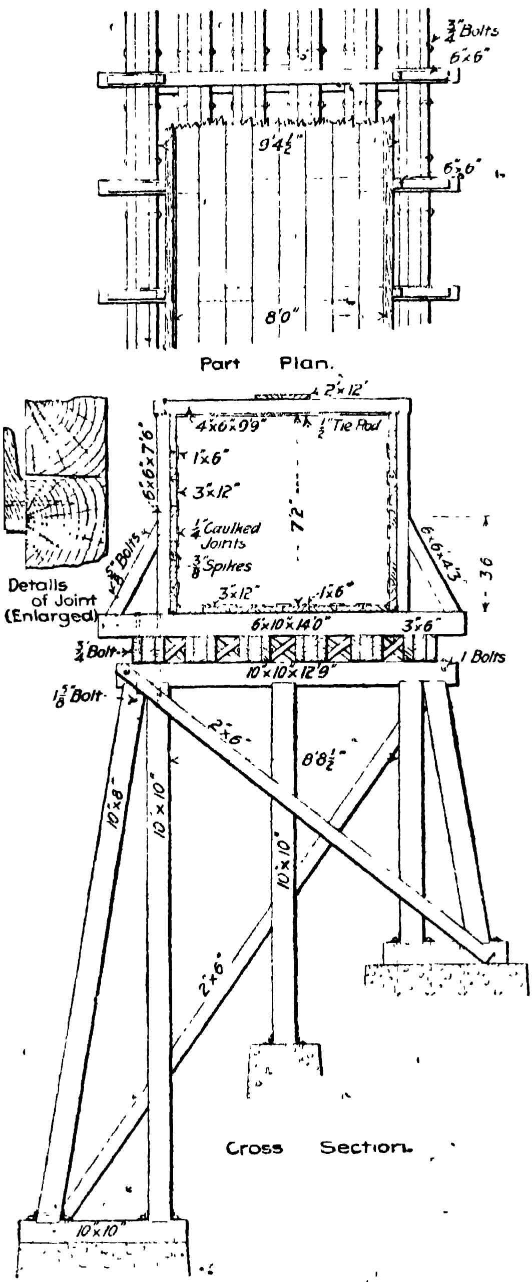


Fig. 71.—Details of Rectangular Wooden Flume.

much less than that of the earthen canal, the overall annual charges of the two types, designed to give the same initial loss of head, are usually not very different.*

Where a canal is cut in the side of a hill, precautions must be taken to prevent debris being carried in during heavy rains. Also spillways should be provided at suitable points, for disposing of any surplus storm water.

72. **Flumes** are used instead of canals where the ground presents special difficulties, either due to unsuitability for excavation or to long detours necessary for crossing ravines. They may be of wood, steel, or concrete, and of rectangular or semicircular section.

Wooden flumes are useful in situations where timber is cheap. They are especially useful for temporary work, but are expensive in the long run owing to the high cost of upkeep.

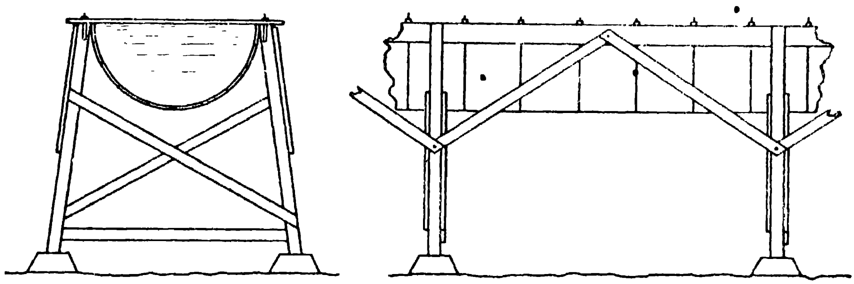


Fig 72 -- Semicircular Wooden Flume

Fig. 71 shows a typical box-flume section. By stiffening the external side braces, the top braces may be eliminated. This, while dearer, renders the flume easier to clean.

Fig. 72 shows a semicircular wood-stave flume. The staves are supported by steel rods, the ends of which are held by stringers. The ends of the rods are screwed, and by tightening up the nuts the staves may be drawn closely together and any cracks closed up. The wooden bents carrying a wooden flume should rest upon concrete or rock, and all contact between wood and earth should be prevented. Otherwise decay is apt to be rapid.

Steel flumes are almost invariably of semicircular section. The plates are usually rolled with a bead at each end so as to form an interlocking joint. Watertightness is ensured by means of a curved rod fitting the outside of the groove, and a curved bevelled bar on the inside. They are supported either on timber or angle-iron bents, in a similar manner to the semicircular wooden flume.

The metal sheets should be galvanized, and should be coated with tar paint or similar preservative. If the water carries much sand or gravel, any such coating is quickly worn away, and to prevent this a settling basin should be provided at the head of the flume.

* See *Engineering News*, 5th Dec., 1906, p. 620, and 27th March, 1913, p. 619.

Reinforced-concrete flumes are much less expensive in upkeep than those of wooden or steel construction, and are coming into more general

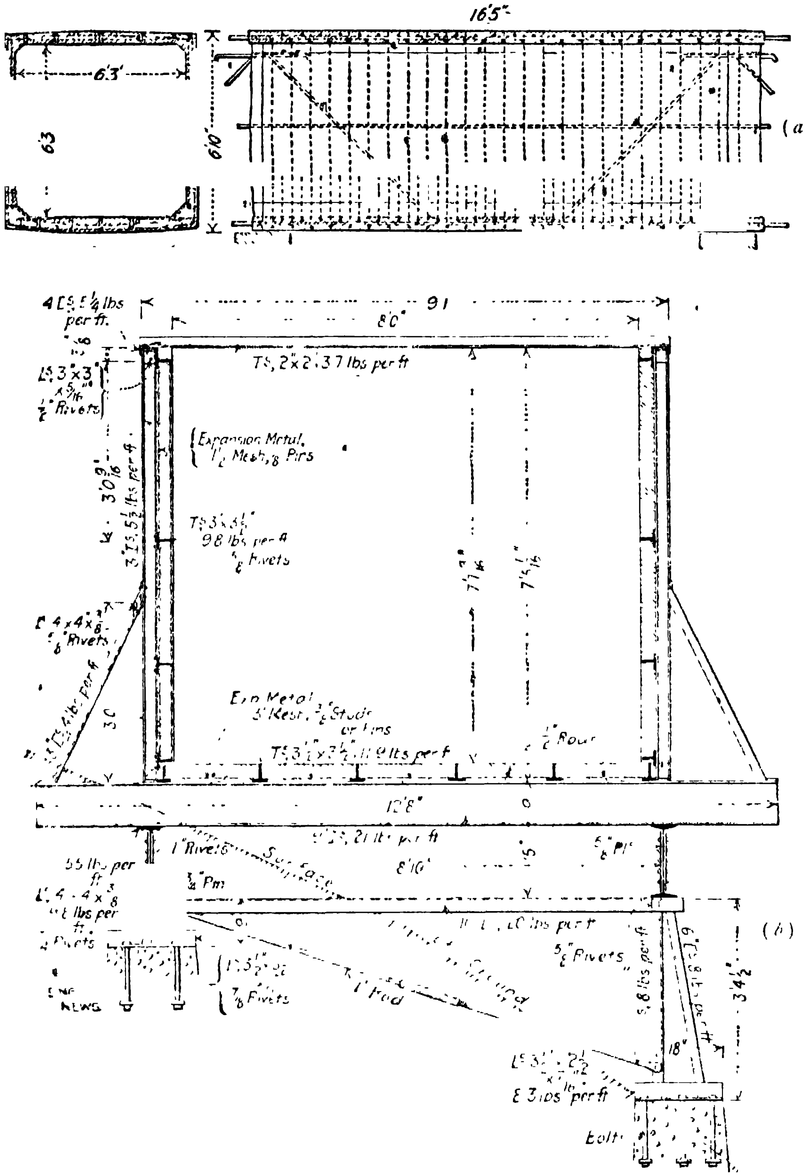


Fig. 73 Types of Reinforced-concrete Flume

use. Fig. 73*a* shows a typical design of a closed concrete flume, while fig. 73*b* * shows an open top-braced flume.

73. Drainage Crossings. Wherever a canal or flume location cuts

* *Engineering News*, 24th Dec., 1908, p. 705

across a natural drainage line, it is necessary to make some provision for dealing with the drainage water. Usually drains or culverts are provided of sufficient capacity to carry away the water without danger of its banking up behind the flume. Where the amount of water available from this source is appreciable, it may be advisable to utilize as much as possible in order to augment the reservoir supply. As such drainage water usually carries considerable sediment, a settling basin must be provided with a bottom opening for sluicing sediment. The water from this basin is fed into the canal or flume over a weir, while a spillway must be provided at some convenient part of the flume, to enable any water, in excess of the capacity of the flume, to be discharged.

Fig. 74 shows the automatic arrangement used for discharging the surplus water from the side streams in the Kinlochleven installation.*

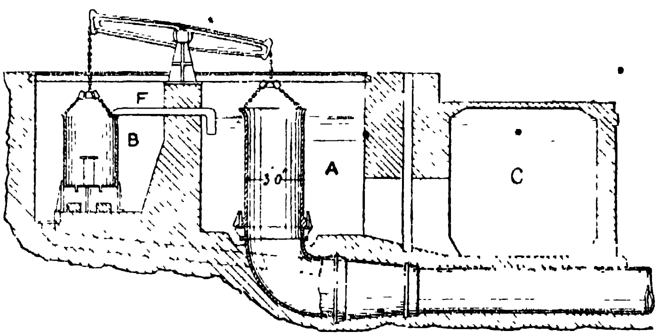


Fig. 74.—Catchwater for Kinlochleven Installation.

Here each side stream is dammed by a concrete dam between 50 and 60 ft. long, whose top surface forms a spillway capable of taking the spate discharge of the stream. The small reservoir formed by each dam serves as a settling basin, and is sluiced at intervals through a sluice gate in the dam. The water from each basin is conveyed by a short concrete channel to a delivery chamber, where it is discharged over a weir into a forebay chamber A, alongside the main conduit C. This chamber is connected to the conduit through a series of five submerged openings, 2 ft. square, and contains two balanced cylinder gate valves as shown. These valves are opened by the filling of a balance bucket, B, with water through a feed pipe F of flat section, when the level in the conduit attains its maximum permissible height. Each bucket is provided with two 1½-in. cocks in the bottom, which can be adjusted by trial. When the inflow exceeds the discharge from these cocks, water accumulates in the bucket, and when the bucket is half full the beam is equally poised. Any further weight in the bucket then opens the gate valves, which when fully open are capable of taking the whole discharge of the side stream.

74. Tunnels are often necessary in a hydro-electric scheme, in order to convey the water through an intervening ridge, or to avoid a long detour.

* *Proc. Inst. C. E.*, 1911–2, Pt. I, p. 49.

Where either a tunnel or a detour is possible, the relative capital costs and maintenance charges determine the most advantageous system. The maintenance costs in a tunnel are usually very low, but the first cost is high. Tunnels may be either of the pressure type or non-pressure type. They are usually lined with concrete to reduce friction losses, and, in a pervious structure, to prevent leakage. For very high pressures a steel lining may be used. While a circular section is most efficient from a hydraulic point of view, a horseshoe section is more common as being more easily excavated, unless some form of tunnel-boring machine is used.

75. Pipe Lines.—Except where the head is low, and the turbine is fed directly from the open forebay, the final stage in conveying the water to the turbines is accomplished by a pipe line constructed of steel plate, cast iron, wood stave, or reinforced concrete. This pipe line is known variously as the power conduit, the penstock, or the supply pipe.

The hydraulics of pipe flow have been discussed in Chapter V, and the general considerations governing the location of the pipe line are considered in Chapter X. Generally the location should be chosen so that the pipe lies below the hydraulic gradient under all conditions. Special care should be taken to ensure this under increasing loads, where inertia effects reduce the pressure below normal. This is particularly important in large steel pipes, which are quite unfitted to withstand any appreciable negative internal pressure.

The size of the pipe is governed mainly by the permissible friction loss, with due consideration of the maximum velocities permissible for successful speed regulation. In practice the friction loss is calculated for a series of diameters. The most economical pipe size is that in which the sum of the capital cost and the capitalized value of the energy lost through friction is a minimum. Since the wall thickness for a given diameter increases with the head, the greatest economy of material for a given friction loss will be obtained by varying the diameter from the top to the bottom of the pipe, the greatest diameter being at the top. While a continuous variation is not practicable, an approximation to it is often obtained by dividing the pipe length into three or more sections of different diameters.

For welded pipes with muff joints, in order to preserve uniformity of joints in each section, the external diameter is often kept the same in each section, the thickness and therefore the internal diameter varying with the pressure.

Wood-stave Pipes.—These have found an increasing use in connection with water-power work, particularly in newer countries with good native timbers.

Pipes for very small installations can be machine-made in sections, with spigot and socket joints, very similar to ordinary cast-iron pipes, but this type is not used to any appreciable extent.

The continuous wood-stave pipe is the more ordinary type. This is illustrated in fig. 75 *a* and *b* which shows the 13·5-ft. pipe line of the recent extension to the Ontario Power Company's plant at Niagara. It

Fig. 757

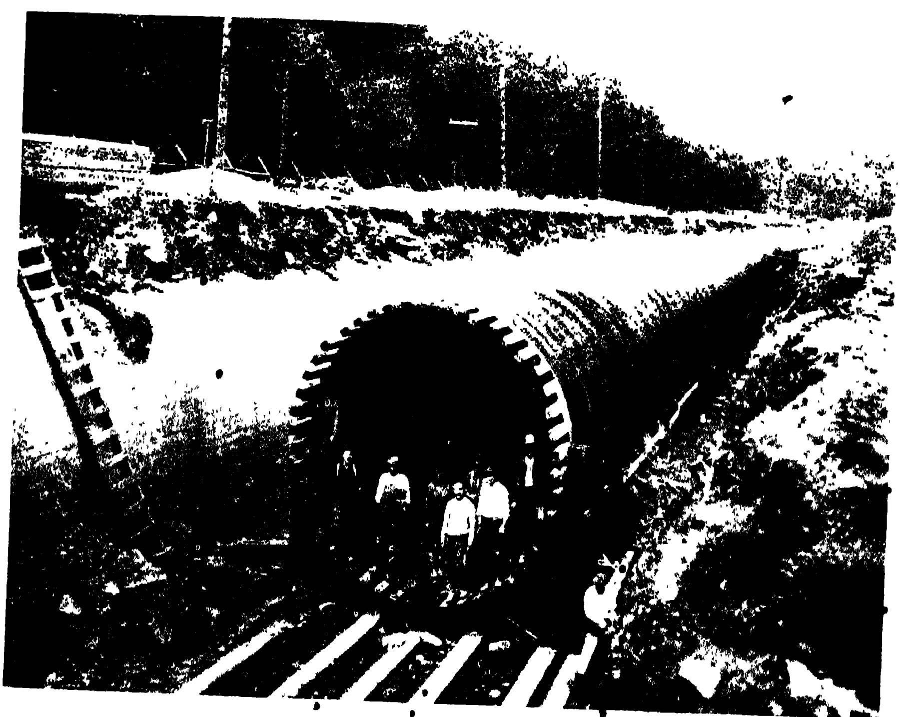


Fig. 756

Wood Stave Pipe Line, Ontario Power Co., Niagara

is built up of staves usually from 6 to 8 in. broad, each stave having radial edges and concentric circular faces. The staves are arranged so that the circumferential joints are not continuous. Leakage at these joints is prevented by a thin metal plate driven into a saw-cut on both of the abutting ends, and covering the joint.

The staves are held together by circumferential steel bands, gripping malleable-iron shoes. One end of the band is screwed and provided with a nut to enable it to be tightened. These bands provide the necessary resistance to bursting, and their diameter (which is usually between $\frac{3}{8}$ in. and 1 in.) and their spacing are determined from the maximum pressure to be anticipated at each point in the pipe.

When the pipe is filled the wood swells and places the bands under some initial tension. This may be taken as equivalent to about 100 lb. per square inch in the timber. For large pipes with fairly high pressures this effect may be neglected.

The size and spacing of the bands should be such that the bearing pressure between band and wood does not exceed about 650 lb. per square inch. In computing this, the width of the arc of contact is usually taken as being equal to the radius of the band.

Wood-stave pipes are usually laid above the ground. They require the simplest of foundations. Bed logs roughly adzed to shape are usually sufficient, although for large sizes a frame support is often employed.

No painting is required except for the bands, which can be dipped in hot tar before being placed. The smooth interior gives a very low friction loss, which diminishes slightly with time, instead of increasing as in other types of pipe. No special anchorages are necessary except where the pipes are used on steep slopes, when anchors will be required at sharp changes of grade, and also at points of junction with any steel pipes or other fixed conductors. No expansion joints are required. Any necessary bends are made during the process of erection, each stave in turn being braced back until the bend is built up.

The materials are easily transported, and neither erection nor repair require any great degree of skill. If suitable timber is available the mill can be set up on the spot, and only the bands and shoes need transporting.

As heads and diameters increase, the amount of steel increases, till it becomes comparable with that required for a riveted steel pipe for the same duty, and the wood pipe ceases to present advantages on the score of cost. In general the range of useful heads is from 20 ft. to 200 ft. Such pipes have, however, been used for heads up to 400 ft., where the cost and difficulty of obtaining steel pipes have been abnormally great.

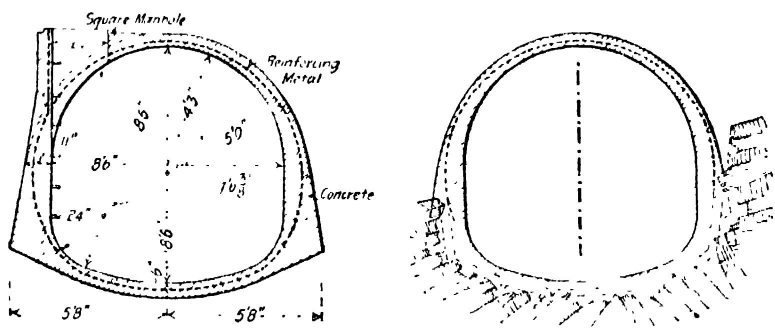
As has already been stated, wooden pipes are usually left exposed and uncoated, their preservation being due to the water slowly penetrating the staves and being evaporated from the surface, thus keeping the fibres saturated. A certain pressure is requisite to produce the necessary penetration. Such pipes are not well suited to heads less than about 20 ft.

Wood-stave pipes have been made in very large sizes, diameters of 10 to 15 ft. not being exceptional.

Cast-iron Pipes are not now often used for power developments, because of the cost and weight for transport. They present no special features. Constructionally, they are not suitable where severe water-hammer action is to be anticipated.

Reinforced-concrete Pipes are now being used to an increasing extent in place of canals and flumes for low-pressure work. Not only are they more permanent and cheap in maintenance, but also, as will be seen in Chapter X, the average effective head on the turbines, in schemes in which the head-water level varies, may be appreciably increased by their use.

As compared with steel pipes the materials are more easily transported, the depreciation charges are much lower, and the friction losses are less.



(*a*) In earth (*b*) In rock

Fig 76 — Typical Reinforced Concrete Pipe Sections

They must, however, be carefully made and properly reinforced, and special care must be taken to obtain reliable foundations. They must be waterproofed, and proper drainage must be provided on the uphill side to protect the foundations against a wash-out. They are specially suitable for heads up to about 100 ft.

Fig. 76 *a* and *b* shows a typical reinforced-concrete pipe section. These pipes are moulded in site, and as the bulk of the materials is usually obtained locally, only the cement and reinforcement require to be transported for any distance. In some cases, especially for small schemes, and where the pipes must be buried, pre-moulded or machine-made concrete pipes with loose-sleeve or spigot-and-socket joints are used.

Steel Pipes are classified as riveted, welded, or lock-bar pipes, according to the method employed in making the longitudinal seams.

Riveted Pipes may be made in a number of ways. The plate may be rolled so that one plate forms the whole circumference, a single longitudinal seam being required. If the pipe is of large diameter the length of the plate has to be used to form the circumference, and the width forms the length of each section of pipe, so that a large number of circumferential seams are required. This entails a somewhat smaller number of rivets on

the whole, but increases the friction losses as compared with the alternative method. Moreover, the plates have either to be bent at the site or the pipe has to be transported in a complete section. Occasionally, but not usually, this method is the better.

The more usual method is to use the length of the plates for the length of the pipe section, three or four plates being used to compose the circumference, as shown in fig. 77. Each plate is bent to a radius slightly greater than that of the average section of the pipe. The joints are lap and single- or double-riveted, as necessitated by the stresses.

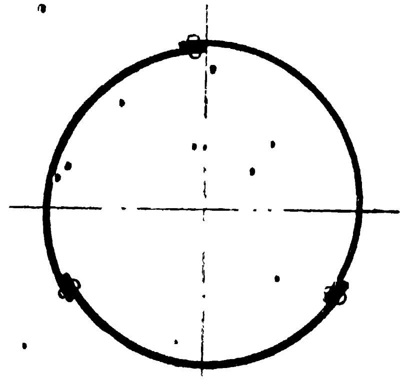


Fig 77

The advantage of this construction is that the plates can be bent, drilled, and scarfed before dispatch, leaving only the riveting to be done in the field. The plates can be bundled for shipment so as to economize space and reduce freights. At the site they can be handled individually if need be, and are riveted up close to their final position. The circumferential joints are riveted after the pipes are actually placed.

The thickness of plate required to resist the bursting pressure is

$$t = \frac{p \times D}{2 \times S \times E}.$$

Where t = thickness of metal in inches.
D = diameter of pipe in inches.
S = working stress in pounds per square inch.
E = efficiency of joint.
p = maximum pressure (including water-hammer) in pounds per square inch.
S = 15,000 for Siemens Martin steel plates.
E = 60 to 65 per cent for single-riveted joints,
= 70 to 75 per cent for double-riveted joints.

The calculated thickness should be increased by $\frac{1}{16}$ in. to allow for corrosion.

For low pressures and large diameters the pipes, if built of this calculated thickness, would be too thin to permit of being handled safely, and to withstand the bending stresses due to the weight of the pipe and water with the supports at a reasonable distance apart. In no case should the thickness be less than $\frac{3}{16}$ in., while for a diameter of 3 ft., $\frac{1}{4}$ in. is the minimum, and for a diameter of 12 ft. and upwards $\frac{1}{2}$ in. is the minimum. In recent large riveted steel mains under low heads, the pipe has been stiffened by longitudinal angle-irons designed to resist the bending moments, and the pipe walls have been correspondingly reduced in thickness.

Circumferential joints are of various kinds. In one method, indicated in fig. 78, alternate sections are made of different diameters so as to fit into each other, and are connected by a single row of rivets. Angle-iron rings welded up and riveted on to the pipe are sometimes used for small pipes. In this case each length is completed in the shop before dispatch. Fig. 79 shows what is known as the "bump" joint. Here the rivets of

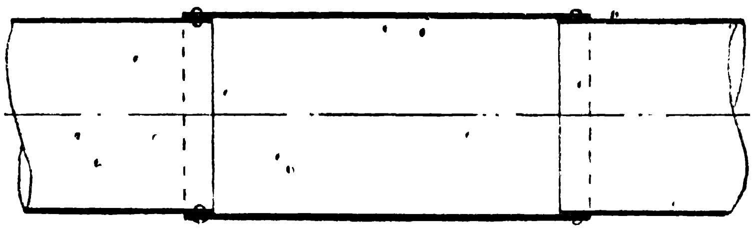


Fig. 78 --Riveted Steel Pipe

the circumferential joint are sunk with a view of reducing friction. It is doubtful, however, whether the gain is appreciable, while this type of joint is expensive.

Welded Pipes.—In welded pipes the longitudinal joint is made by lapping over and welding together the edges of the bent plate, the weld being worked until the thickness is reduced to that of the body of the

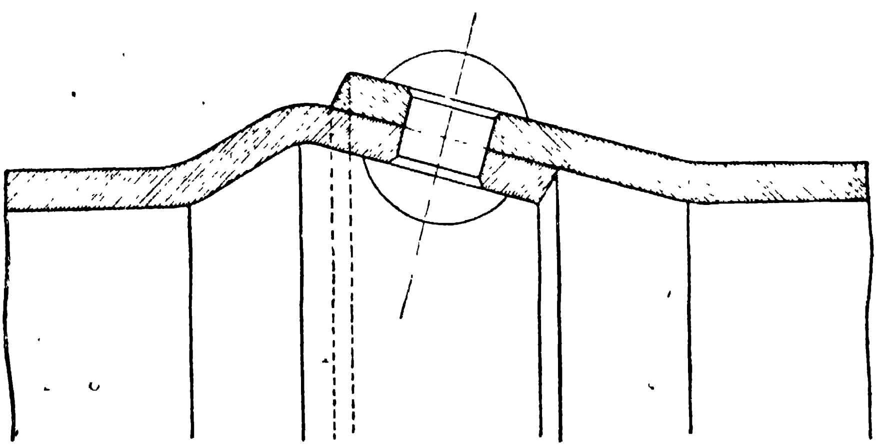


Fig. 79 --"Bump" joint for Steel Pipe

plate. The pipe is afterwards annealed to remove internal stresses. This method of welding is unsuitable for thicknesses exceeding about $1\frac{1}{4}$ in. For greater thicknesses the two edges of the plate are brought together, and a separate wedge-shaped bar is inserted to form the weld. In this way plates up to about $1\frac{3}{4}$ in. thick can be welded. The efficiency of a welded joint may be taken as between 90 and 95 per cent. Owing to the greater thicknesses which are possible, welded pipes are suitable for heads and diameters greater than those possible with riveted pipes, while

owing to the absence of rivets the friction losses are lower. Siemens Martin steel, with a tensile strength of approximately 28 tons per square inch and an elongation of 20 to 25 per cent is the most suitable material.

In several modern installations in which large diameters and high heads

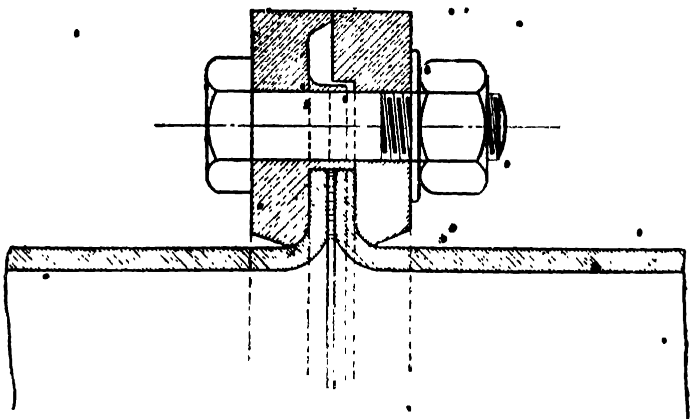


Fig 80 Type of Circumferential Joint for Welded Steel Pipe

have been combined, reinforced welded pipes have been used. In this construction a number of welded steel rings are shrunk on to the outside of the pipe. This enables a larger diameter to be used without exceeding the limit of pipe thickness capable of being reliably welded.

For extremely high heads, where only comparatively small diameters

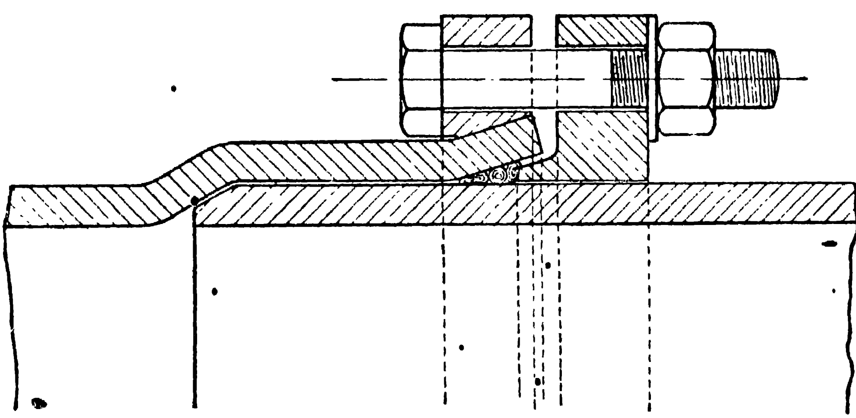


Fig 81 —Single Muff Joint for Welded Steel Pipe

are necessary, solid drawn pipes may be used. In the Fully installation, in Switzerland, in which the statical head is 5412 ft., corresponding to a pressure of 2360 lb. per square inch, welded pipes are used where the thickness is less than $1\frac{3}{8}$ in. In the lower portion of the pipe line, where the thickness varies from $1\frac{3}{8}$ in. to $1\frac{3}{4}$ in., solid drawn steel pipes are used. These are made in short lengths, which are welded together with a

circumferential weld into lengths of about 20 ft. The diameter varies from 23·6 in. at the top to 19·7 in. at the bottom of the pipe line.

The circumferential joints of a welded pipe may be formed in a variety of ways. Flanges can be welded on, or loose flanges can be placed on the pipe and the ends afterwards rolled out as shown in fig. 80.

The more usual joint is that known as the single muff, shown in fig. 81. One end of each pipe is rolled out to form a socket for receiving the plain end of the adjacent pipe. It is further opened out to form a space into

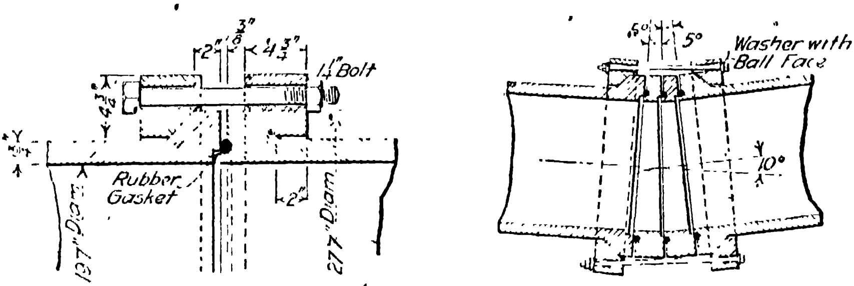


Fig. 82.—Pipe joints used on Lac Fully high-head plant

which the packing for making the joint watertight can be placed. A loose cast-steel flange ring fits against the shoulder thus formed, and the packing is tightened down by a loose cast-steel nose ring sliding on the pipe next above as shown. The packing is usually of hemp soaked in oil and mixed with chalk or other hardening material. English practice favours square-ended nose rings with a packing space having parallel sides. Continental practice favours wedge ends and a conical space, which necessitate a firmer type of packing. This joint is flexible, in that it can bend about 1° in any direction, so that it allows for any slight errors in survey or setting out, or even slight subsidences. It is suitable for very high heads up to 2000 ft. If necessary the joint can be repacked, without disturbing the pipe line, by slacking back the nose ring.

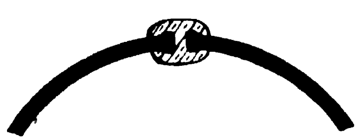
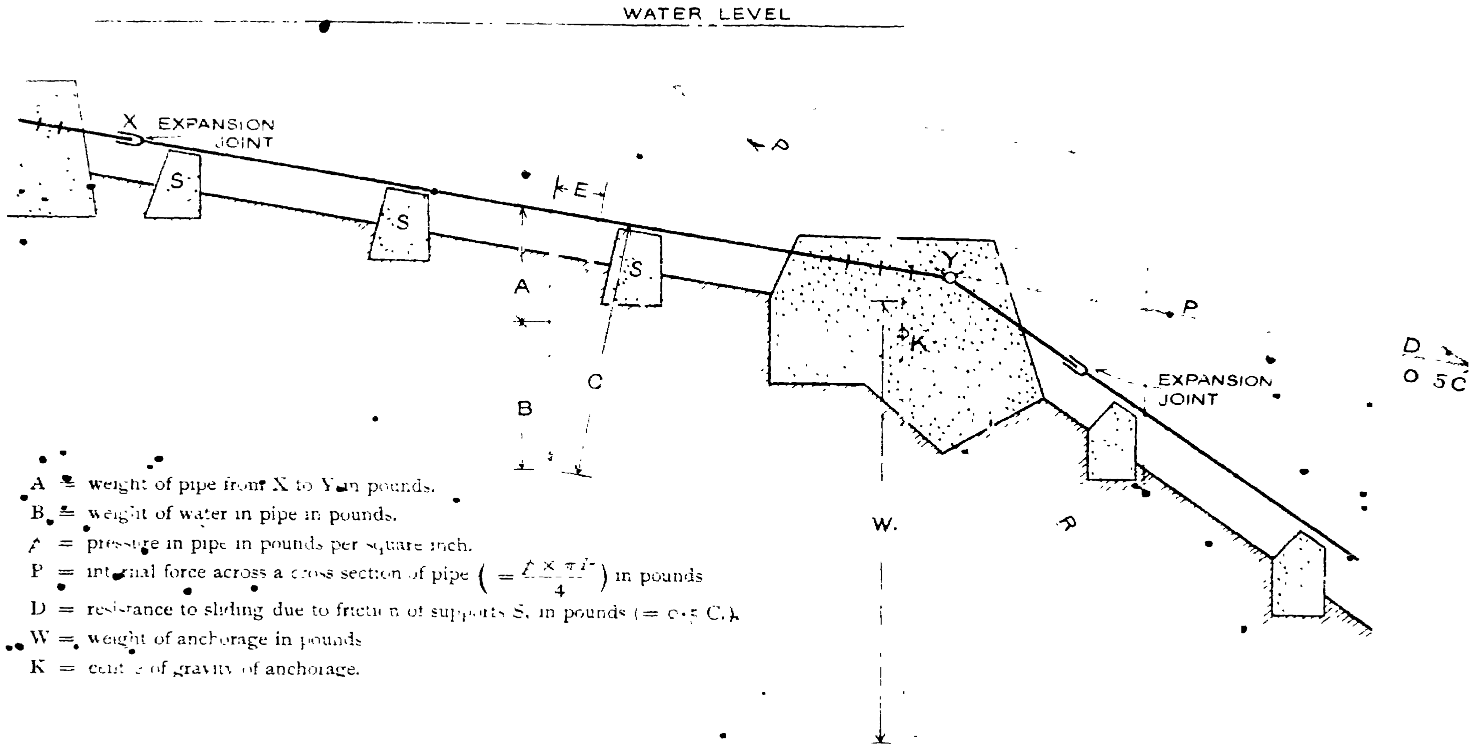


Fig. 83.—Lock-bar Joint

Fig. 82 shows the type of joint adopted for the lower section of the pipe line of the Fully Hydro-electric Station (p. 123). Each end of a pipe section is provided with conical flanges, which are pulled together by bolts passing through two loose steel rings. In order to get these rings on to the pipes, each is made in two halves which are bolted together. At points of divergence from the straight, the deviation is produced by the insertion of one or more wedge-shaped joint rings as shown in the sketch.

Among other types of pipe the spiral riveted pipe should be mentioned. This is useful only for comparatively low pressures and small diameters.

The lock-bar joint (fig. 83) has been used to a limited extent. The friction loss is relatively small. The disadvantages are that such pipes must be transported complete to the site, and that the method of con-



A = weight of pipe from X to Y in pounds.
B = weight of water in pipe in pounds.
p = pressure in pipe in pounds per square inch.
P = internal force across a cross section of pipe $\left(= \frac{p \times \pi d^2}{4}\right)$ in pounds
D = resistance to sliding due to friction of supports S, in pounds (= 0·5 C.).
W = weight of anchorage in pounds
K = centre of gravity of anchorage.

FIG. 84.—DIAGRAM OF FORCES ON PIPE ANCHORAGE

struction is only suitable for a small range of thicknesses. The efficiency of the joint is about 90 per cent.

76. Pipe Supports.—Pipes may be laid underground, if this course is necessary to preserve the amenities of a district, and in this case no special foundations or anchorages are required except on very steep slopes, and no expansion joints are necessary. Since such pipes cannot be inspected and recoated externally, the deterioration is more rapid than in a pipe laid above the surface, unless they are laid in concrete, and the exposed pipe is to be preferred wherever possible.

When laid above the surface, the pipes are supported on main anchorage blocks at each horizontal or vertical bend, and on lighter conorete saddles at intermediate points. The supports should be founded if possible on bed-rock, and in setting out the pipe track the rock line should be followed as far as possible. Occasionally pipe lines have to be laid over ground too deep for excavation to solid rock. Sufficiently stable foundations may then be obtained by the use of large blocks of mass concrete, although there is always an element of risk if the foundation is not on solid rock. With welded pipes the intermediate saddles are placed near each joint. With riveted pipes sufficient clearance is to be provided between saddle and joint to allow of expansion and contraction without fouling. A support at alternate joints is sufficient, provided the pipe is designed to withstand the bending stresses involved.

Adhesion between the pipe and its supports may be prevented by protecting the pipe with greased paper before the concrete is poured. Any such adhesion tends to crack the supports when expansion and contraction take place.

The Main Anchorages must be of sufficient weight to ensure that the line of action of the resultant of all the forces acting above the base of the block shall fall well within the area of the base. In addition to the weight of the block, these forces include the pressures in the pipe (including any water-hammer pressure) acting in the direction of the pipes above and below the anchorage, the weight of the pipe and the included water above and below the anchorage to the nearest expansion joints, and the frictional forces which may be set up at the intervening saddles by expansion and contraction. Fig. 84 shows the diagram of forces in the case of a welded pipe with an expansion joint immediately below the anchorage. In order to transfer these forces to the concrete block, angle flanges are riveted or welded to the bend. If necessary, cast-iron collars are provided to distribute the pressure, and if the upward force on the pipe is considerable, this must be anchored down to the block.

If long straight lengths occur, it is usual to subdivide these into lengths of 200 yd. or less by putting in intermediate anchorages of much the same type as, but lighter than, the main anchorage. This is necessary both to reduce the length expanding and contracting as one unit, and to reduce the forces on the main anchor blocks due to the friction on the supports.

77. Expansion Joints. For welded pipes these are made very

simply by extending the depth of the socket of the muff joint of fig. 81, or as in fig. 85. For riveted pipes a similar type of sliding joint may be employed, or a bellows joint as shown in fig. 86. A more flexible joint may be obtained by modifying this design, and by using flat disk sides separated by a wider ring.

The proper location of the expansion joint for welded pipes with muff joints is immediately below each main and intermediate anchor block. For riveted joints the best position is midway between anchor blocks, so that the movement of the individual lengths of pipe due to expansion is halved.

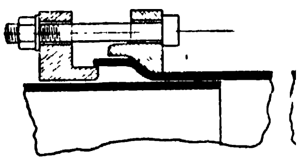


Fig 85 --Expansion Joint for Welded Pipe

In American practice expansion joints are often omitted, even where the pipes are not buried. While their omission does not necessarily lead to excessive stresses in the pipe walls so long as the pipe is full, it does tend to increase the loading of the anchor blocks, and it is probable that the use of expansion joints will become more general in future.

78. Accessories to Pipe Lines.--Ample provision of manholes should be made for inspection and painting the interior. Such work is not particularly pleasant, and proper facilities must be provided to secure thoroughness. Moreover, most protective paints give off oppressive vapours, and as much air as possible should be allowed to circulate through the pipe while it is being coated.

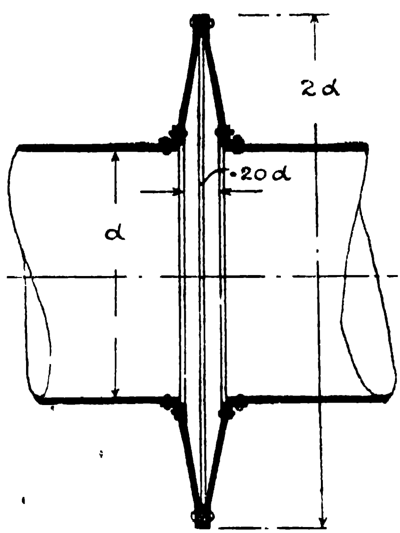


Fig 86 Expansion Joint for Riveted Pipe

An open-vent pipe carried above head-water level should be provided close to the entrance of the pipe line, to release any trapped air and to prevent a vacuum being formed by the rapid closure of the head gate or valve, or by a failure of the lower part of the pipe.

If, for any purpose, valves are installed in the pipe line at intermediate points, as usually occurs when the number of pipes changes at such intermediate points, an air admission valve should be fitted below each main valve. This usually consists of a spring loaded valve opening inwards, and admitting air to the pipe if, owing to the closure of the main valve, the pressure in the pipe below the valve becomes less than atmospheric.

Where a pipe line is taken over a ridge, an air-vent valve must be fitted at the highest point to allow the release of imprisoned air each time the

pipe is filled. This usually takes the form of a ball float valve which falls as air accumulates, and allows it to escape. Fig. 87 shows the type fitted to the 42-in. supply mains of the Kinlochleven Power Works.* The float, working between guides, carries an air valve spindle at its upper end, and the upper part of the valve casing serves as an air vessel to reduce shocks in the pipe. Such valves also serve to admit air to the pipe line and to prevent the formation of a vacuum in the case of a fracture at some lower point in the line.

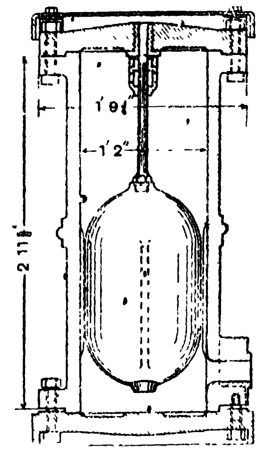


Fig. 87. Air Admission Valve

Proper arrangements must be made for emptying the pipe. If it has a continuous slope from forebay to turbine, with no intervening valves, no special provision is required, but if it has low points, or is divided into sections by valves, a drain-pipe and valve must be provided, at the lowest point of each section.

79. Relief Valves.—In order to prevent any harmful effect due to pressure surges consequent on the closing of the turbine gates, some form of relief valve is necessary at the lower end of a long pipe line. The best type of relief valve is one controlled by the governor, opening as the gates close, and afterwards closing slowly by its own weight (Art. 99). Differential pressure relief valves of the general type shown in fig. 88, which shows a Lombard valve, are occasionally fitted as an emergency device. Here P is the pipe line with its relief valve V, which is held up to its seat by the water pressure on the piston R. The space below this piston is connected through the pipe A to the waste valve B. This is a balanced valve held up to its seat by the spring S against the pressure of the water in the penstock, which acts through the pipe C on the piston D. If this pressure becomes

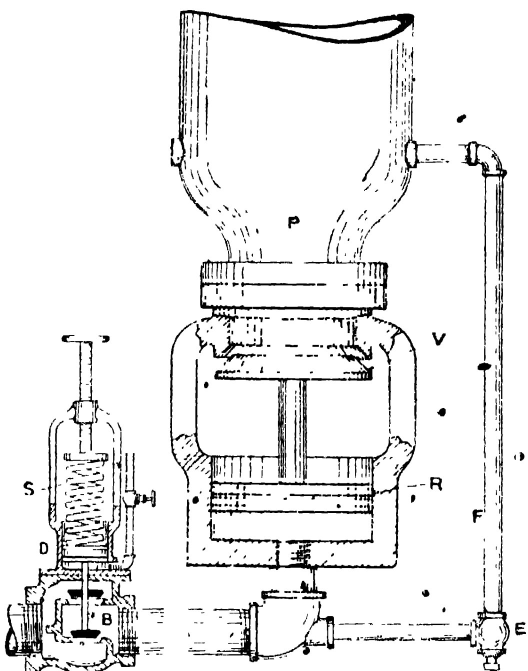


Fig. 88. Lombard Relief Valve

* *Proc. Inst. C. E.*, 1911-2, Vol. 187, Pt. I, p. 128

greater than normal, the valve B is forced open, allowing water to escape and relieving the pressure behind the piston R. The relief valve V then opens. When the pressure in the pipe line falls to normal the valve B again closes, and the pressure behind the piston R gradually increases and closes the relief valve. The rate of closing can be regulated to prevent surging by suitable adjustment of the throttle valve E on the connecting pipe F.

Spring and weight-loaded relief valves may be used at points immediately above main valves whose rapid closure would give rise to dangerous pressures.

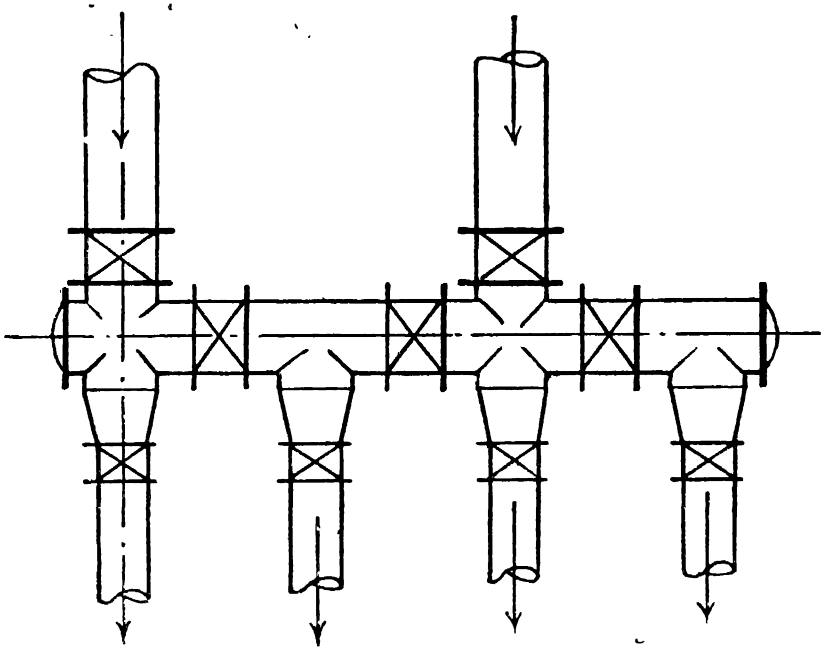


Fig. 89

80. Interconnecting Pipes.—It is not unusual for the number of pipes to be increased at some point between the forebay and the turbines. At such a point the pipes are commonly interconnected. This gives a more uniform distribution of flow through the various pipes, and also enables any one pipe of the upper or lower series to be isolated for inspection or repair without affecting the remainder. Fig. 89 shows a typical case. It will be seen that a valve is required for each of the upper and lower pipes, and it is desirable also to provide a valve for each section of the interconnecting pipe, otherwise any failure in the interconnecting pipe itself may shut down the whole station. The valves may be of the ordinary sluice type, but those at the head of the lower pipes may conveniently be made of the type illustrated in fig. 67, p. 110.

In many cases the pipe lines are interconnected immediately outside

the power house, so that in the event of any one pipe or turbine being out of commission the rest of the plant can be run. The usual plan is to use lengths of pipe, of about the same diameter as the main pipe lines, to connect each adjacent pair of pipes. This involves a good deal of expensive work, since for high pressures the cross pieces must be made of cast steel, and high-pressure valves must be fitted into each pipe line above the interconnecting pipe, and into each section of the latter pipe.

The extra cost must be weighed against the value of the extra degree of reliability obtained. For schemes in which each turbine is not fed by a corresponding pipe, an interconnecting pipe is absolutely necessary. This is usually the case in low-pressure schemes where pipes of a large diameter are possible.

In modern practice, owing to the increased size of the units and the fact that the pipe diameter is often limited by the maximum thickness of plate that can be used, it is usual to feed each turbine by its own pipe in schemes involving high or moderately high heads, and in many modern stations the interconnecting pipe is omitted.

If separately-driven exciter units are installed, their supply may be obtained from special pipe lines. More usually they are fed from a comparatively small bus-pipe placed above the main pipe lines, and drawing a supply from each or all as desired through the connecting valves.

81. Pipe Coatings. In order to prevent rapid deterioration and incrustation of the interior surface of a steel or cast-iron pipe, a protective coating should be applied.* The most effective coating appears to be one of bitumen or pitch in smooth and perfect layers without pinholes. A mixture of 9 parts of pitch to 2 of boiled linseed oil is said to give good results, and also a mixture of natural asphalt with sufficient liquid asphalt to fill the voids in the dry rock. If the pipe is not too large, and is built up in the shops, the coating should be applied by dipping in a bath at a temperature of about 300° F., the pipe being left in the bath for a sufficient time to enable it to attain the same temperature. The first coating should be allowed to cool before the second dipping, which should not be hot enough to melt the first layer.

There are also a number of proprietary paints on the market which may be applied by a brush or sprayer, and which have proved capable of giving good results. These usually have an inflammable medium, and require caution in application.

82. Shipment.—The question of shipment has already been raised in connection with riveted pipes having three or more longitudinal seams, and the method of reducing shipping space by bundling has been indicated. Welded or other pipes built up before dispatch may be nested. As for economy of material the pipe line is usually built up of a series of diameters, a pipe of the smallest diameter may be shipped inside one of the next size, and so on. In this way the weight for a given volume is increased, often with a considerable saving in freightage. The pipes

* See Brown, *Proc. Inst. C. E.*, Vol. 156, p. 1.

composing the nests are separated by wooden wedges, and to allow for the thickness of metal, and to give space for wedging, there should be a difference in diameter of not less than $3\frac{1}{2}$ or 4 in. between one size and the next. Care is to be taken that the weight of each nest does not exceed a reasonable figure, or otherwise penal rates may be incurred for freightage, cranage, &c. It will be seen that the design of a pipe line may require to be modified to a certain extent in order to take advantage of the shipping facilities.

83. Tail Races. The design of the suction or draft tube is discussed in Art. 87. The water emerging from this tube must be discharged into the stream, and in many cases a tail race is necessary for this purpose. If the power station is on the bank of the stream the tail race is formed in the substructure of the station.

The illustrations of Chapter X give a good idea of the arrangement usual in this part of the scheme. The design must ensure that the draft tubes are water-sealed under all conditions, and that the free discharge of water is not impeded in any way. Where vertical draft tubes are used, the depth of water below the lower end of each tube should be at least equal to its outlet diameter. This condition will often necessitate lowering the bottom of the suction pit. The mean velocity in the suction pit should not exceed from $1\frac{1}{2}$ to 2 ft. per second. A velocity of 3 to 4 ft. per second may be allowed in the tail race itself.

It is usual to provide stop-logs at the point of exit from the power station, so that when necessary the suction pits may be isolated for inspection and repair. Where there is a large seasonal variation in the level of the tail water, it may be necessary to place the floor-level of the station so high above the lowest level to which the tail water may fall, as to exceed the permissible suction height, the approximate figures for which are given in Sec. 87, p. 146. In such a case it is necessary to install a weir to maintain a sufficient level in the suction pit. The slots for the stop-logs may conveniently be used for this purpose. If this low-water condition is likely to be frequent, and of long duration, a vertical shaft turbine should be installed if possible, so as to enable the available head to be utilized at all times. This question is discussed more fully in Chapter X.

It is a matter of arrangement as to whether all the units in the power station shall discharge into a common suction pit running the whole length of the power house, or whether they shall discharge into a series of pits each having its own exit at the side of the station. The former method is the cheaper, but suffers from the disadvantage that if repairs have to be executed necessitating the draining of the suction pit, the whole station requires to be shut down.

CHAPTER VIII

Water Turbines

Turbines; impulse and reaction turbines; comparison; constructional details; draft tubes; hydraulics of the reaction turbine; model tests; characteristic curves; specific speeds; limiting size of runners; Pelton wheels; constructional details; hydraulics of speed regulation; run-away speeds; selection of turbines.

84. Modern turbines may be divided into two classes, Impulse and Reaction turbines. Of the former the Pelton wheel, and of the latter the Francis turbine, or one of its modifications, are the only types used in recent important installations.

In an *Impulse turbine* the whole head of the supply water is converted into kinetic energy before the wheel is reached. The water leaves the nozzle or nozzles in one or more high-velocity jets which are exposed to the pressure (usually atmospheric) obtaining in the turbine casing. It then impinges on a series of buckets carried by the wheel, and in virtue of the change of direction and hence of tangential momentum produced by these buckets, exerts a driving force and so does work on the shaft. Its direction is freely deviated by the buckets, and its pressure remains uniform during its passage through the turbine.

The *Pressure* or *Reaction turbine* consists essentially of a wheel or runner provided with vanes into which water is directed over the whole periphery by a series of guide vanes. The water on leaving these guide vanes is under pressure, and supplies energy partly in the kinetic and partly in the pressure form. In its passage through the runner the pressure energy is utilized in increasing the relative velocity of flow between the vanes, and the water finally leaves the runner at the pressure obtaining in the discharge pipe or draft tube.

In the earliest of these turbines, the Fourneyron, the guide vanes were inside the runner, forming an outward flow turbine (fig. 90). This was followed by the Jonval turbine, in which the guide vanes are above the runner and the water flows axially into and through the wheel, giving an axial-flow turbine (fig. 91). Both these types have been to all intents obsolete for some years, and the Francis or inward-flow turbine, in which the guide vanes surround the outer periphery of the runner, has been in general use. In the earlier Francis turbines the discharge was also radially inwards, but in modern turbines, in order to obtain a larger discharge area with a given diameter of runner, the form of the buckets has been modified so as to give a discharge in a direction which is more or less parallel to the axis of the turbine. In the most recent turbines for low-head plants, the design is indeed tending to a turbine which, as regards the runner, is essentially of the Jonval or axial-flow type (fig. 98, p. 140). Inward-flow guide

vanes are used, but there are signs that even in this respect the design of low-head turbines is tending to revert to a modified Jonval type. At the present moment low-head high-speed turbine design is in a state of rapid development, and it is probable that the standard of a few years hence will differ very appreciably from the Francis type.

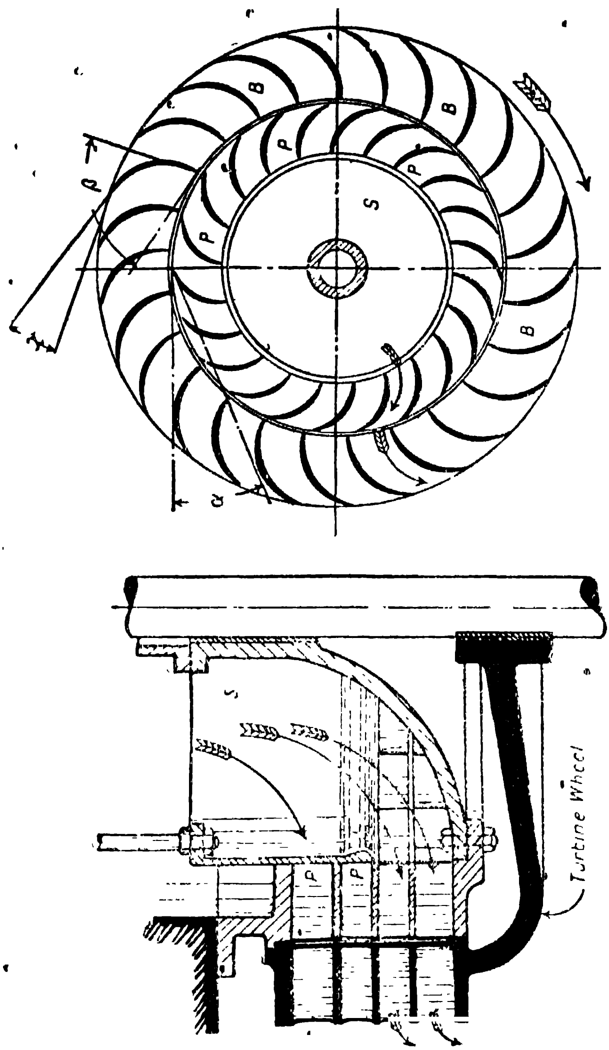


Fig. 90.—Horizontal and Vertical Section through Receiver and Guide Vanes of Fourneyron Turbine with Inside Cylinder Gate Regulation

85. General Comparison of Impulse and Reaction Turbines.— The peripheral velocity of a Pelton wheel for maximum efficiency is slightly less than one-half the spouting velocity of the jet (usually approximately $\cdot 46\sqrt{2gH}$, where H is the head), while that of the reaction turbine varies from about $\cdot 65\sqrt{2gH}$ to $1\cdot 05\sqrt{2gH}$, depending on the design. Because of this, the Pelton wheel is well adapted for very high heads, which may

then be utilized with moderate speeds of rotation. On the other hand, the relatively high speed of the reaction turbine enables reasonably high rotative speeds to be obtained with low heads.

The Pelton wheel cannot well be designed to utilize efficiently more than two jets on a single wheel, and as the maximum practicable jet diameter is about 12 in., the volume of water which can be handled and the output of the turbine become small under low heads. The reaction turbine with its full peripheral admission is well adapted for large volumes. It is not suited for small powers under high heads, since the volume of water is small, the waterways are of very small sectional area and easily become choked by floating debris, and the fluid friction losses become relatively high.

The Pelton wheel is not well adapted to be used with a suction or draft tube, and, where the tail-race level varies appreciably, must be installed above the highest tail-water level with some sacrifice of head. The reaction turbine lends itself readily to this construction, and has the further advantage that it may be drowned without loss of efficiency. The efficiency of the reaction turbine is not so sensitive to changes of head as that of the Pelton wheel, and since the percentage variation in head is usually greater in low-head than in high-head plants, this is another reason why the Pelton wheel is not well adapted for low heads.

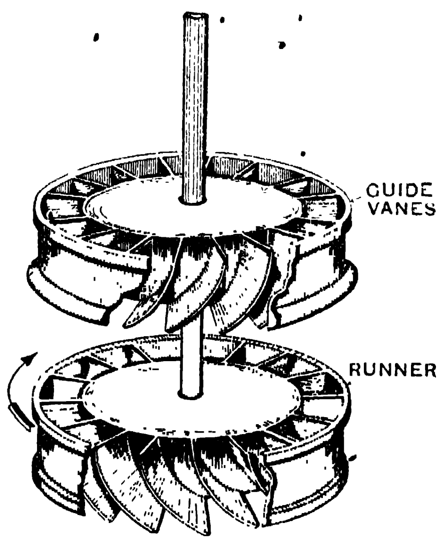


Fig. 91.—Guide Vanes and Runner of Jonval Turb

If operated under constant head and constant speed, the efficiency of the Pelton wheel does not fall off so rapidly at part loads as that of the reaction turbine. On the other hand, the modern reaction turbine has a slightly higher full-load efficiency, so that the average efficiency from half to full load is sensibly the same in a well-designed machine of either type. The following table shows typical values of the part-load efficiencies of modern turbines of both types of large size, installed under equally favourable conditions.

Proportion of maximum discharge ..	·2	·3	·4	·5	·6	·7	·8	·9	1·0
Efficiency of reaction turbine	·60	·70	·75	·79	·82	·87	·90	·91	·89
Efficiency of Pelton wheel	·70	·78	·82	·83	·84	·85	·86	·85	·83

While efficiencies as high as 93 per cent are on record for large reaction turbines, such values can only be attained by the most careful attention to the design not only of the turbine but also of its setting, and in general the full load efficiency does not greatly exceed 85 per cent in the case of a large reaction turbine, and 80 per cent in the case of a large Pelton wheel. The possibilities of accurate speed regulation are about equal in the two types.

For large units the reaction turbine is generally preferable for heads up to 400 ft. For heads above 750 ft. the Pelton wheel is more suitable, while between these limits the choice depends largely upon local circumstances and on the power required. The greater simplicity and accessibility of the parts requiring replacement due to natural wear and tear

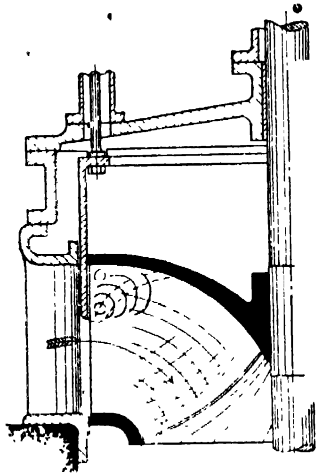


Fig. 92.—Cylinder Gate for Francis Turbine

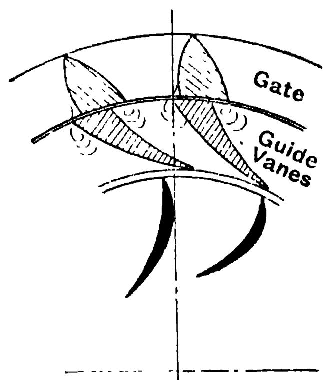


Fig. 93.—Register Gate

renders the Pelton wheel more suitable when the supply is taken from a stream carrying an appreciable amount of grit or silt in suspension.

86. The Reaction Turbine: Constructional Details.—The supply of water to the runner of the reaction turbine is regulated either by a cylinder gate, a register gate, or by pivoted guide vanes. The cylinder gate (fig. 92) consists of a plain cylinder sliding axially between the guide vanes, which are fixed, and the runner. Its axial position is regulated by the governor. The register gate (fig. 93) consists of a cylinder carrying appropriate waterways, and capable of rotation about the axis of the turbine. It is usually fitted outside the guide-vane ring.

With either type excessive eddy formation is produced at part gate, giving low part-load efficiencies. While the register gate is practically obsolete, the cylinder gate is still used in small plants having a fairly constant load, and where low part-gate efficiency is unimportant. It is cheap and not so easily deranged as the wicket gate, as the system of pivoted guide vanes (fig. 94) is termed. The latter arrangement is, however, the only one now used in important installations.

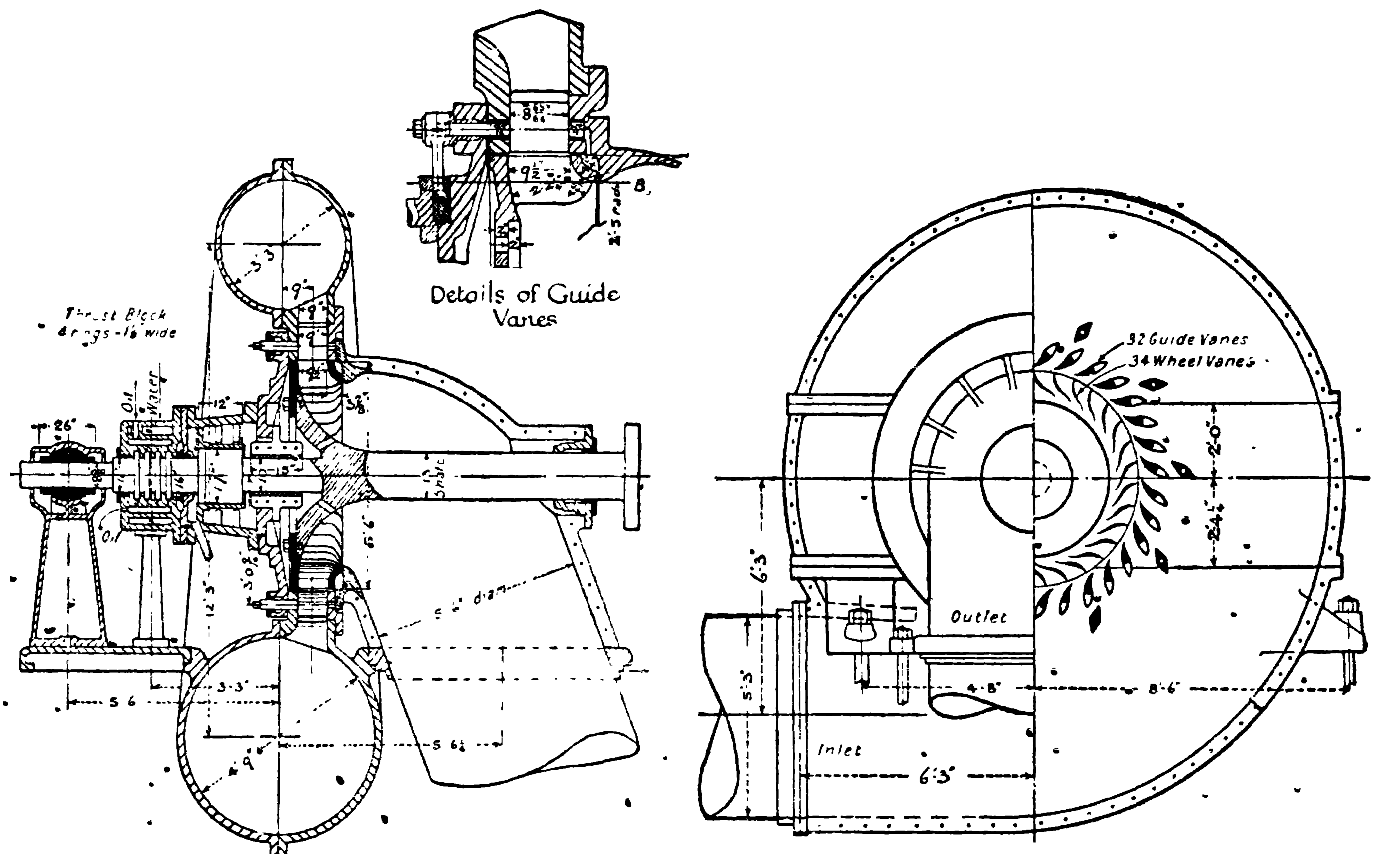


Fig 94.—Section of Francis Turbine for Snoqualmie Falls

In a low-head installation the turbine may be erected in the open forebay as shown in fig. 143, p. 207. This method has the disadvantage that the guide-vane mechanism is submerged, and cannot be inspected or repaired without draining the wheel pit, and in most recent important medium- and low-head installations the guide-vane ring is surrounded by a spiral volute chamber, from which the pressure water is delivered with uniform velocity around the entire periphery of the guide ring.

The velocity in this volute ranges from $\cdot 15\sqrt{2gH}$ to $\cdot 25\sqrt{2gH}$, the higher value applying to low-head plants, and the lower to heads exceeding 300 ft.

For heads not exceeding about 100 ft., modern practice has favoured the moulding of the volute chamber in the concrete of the substructure (fig. 95).* Such a construction has many advantages. As compared with an installation in an open forebay it is more efficient, and only the guide vanes and the runner are submerged. The cross section of the volute may be made of any required shape, while the concrete lends itself to a smooth finish and to the formation of easy curves, both of which tend to efficiency in operation.

For higher heads, considerations of strength necessitate a metal casing, which may be of cylindrical section, but which, for single-runner machines, is of spiral volute form (fig. 94). This may be of steel plate, cast iron, or cast steel. Owing to the risk of flaws in the casting and of its unsuitability for withstanding sudden shocks, cast iron is not very suitable for large casings subject to high heads and liable to water-hammer shocks.

The two sides of the casing are tied together by a series of spacers surrounding the guide-vane ring (fig. 94). These are usually shaped so as to guide the water into the guide vanes, and form what is termed the speed ring. The speed ring is also fitted where the volute is moulded in the concrete substructure, and then consists of two speed-ring crowns connected by spacers. In modern vertical shaft units of this type the load is transmitted vertically downwards through the concrete below the generator to the speed ring, and through its vanes or spacers, which act as columns, to the base of the substructure.

Guide Vanes.—The guide vanes or gates are commonly made of cast steel. The stems may be cast in one piece with the vanes, or, in large units, may be keyed to the vanes so as to facilitate the removal of worn vanes. Under very high heads, where the water carries an appreciable amount of grit, bronze guide vanes are advantageous for resisting erosion.

The stems project through stuffing boxes in the turbine casing. Each stem carries a lever which is coupled to a common regulating ring concentric with the turbine shaft (fig. 94), whose position is regulated by the governing mechanism, so that all the guide vanes are opened or closed simultaneously.

The gate stems should be strong enough to resist the stress which

* By courtesy of the Cramp Shipbuilding Co., Philadelphia.

would be produced in the case of an obstruction between two vanes with the full effort of the governor concentrated upon them. The links between the levers and the regulating ring should be the weakest part of the system.

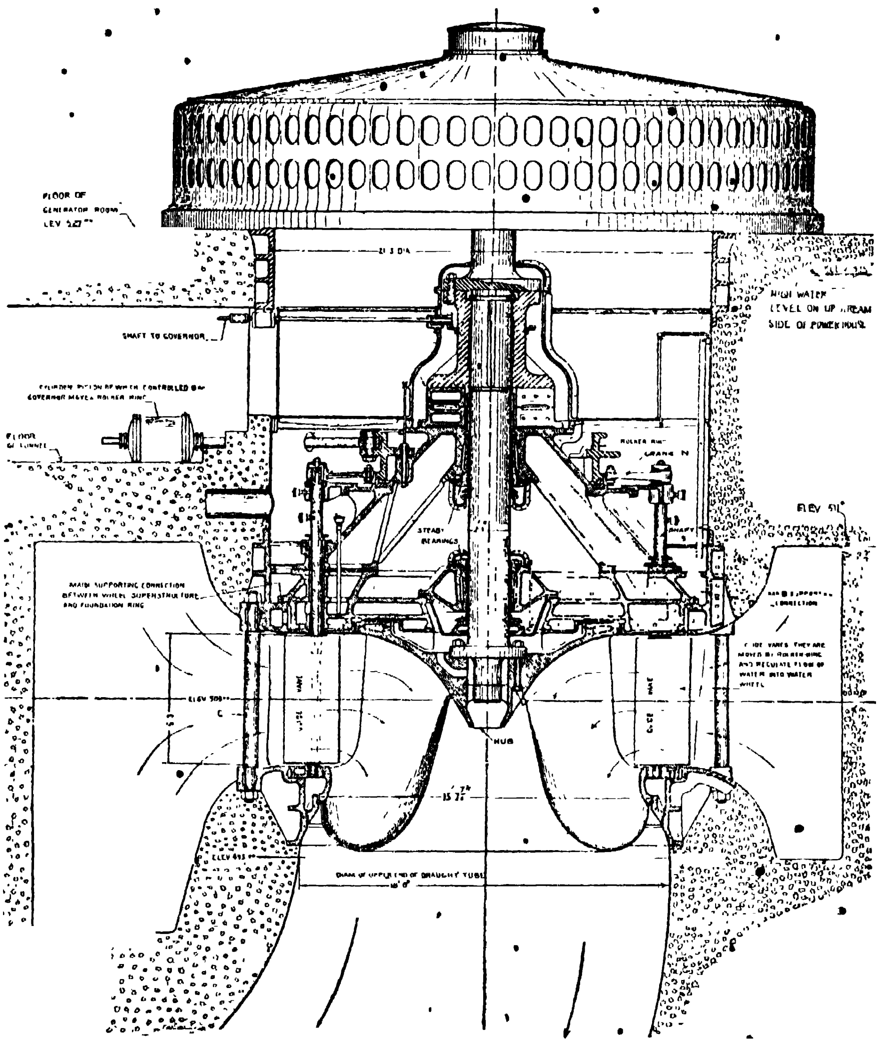


Fig. 95 —Keokuk Turbines of Mississippi River Power Company, Overload capacity, 13,500 h p

Turbine Runners.— For turbines of moderate size under low and moderate heads, the iron runner, cast in a single piece, is general. In order to facilitate shipment and to provide greater assurance of sound castings in very large machines, the runner is usually cast in four quadrants,

which are tied together by a heavy cast-steel crown at the top, and by a cast-steel ring which is a force fit around the lower band of the runner (fig. 96).*

For slow-speed runners of the types shown in fig. 97 *b* and *c*, steel-plate vanes may be used with advantage. These are pressed or hammered to shape against a former, and are cast into the turbine crowns. They have the advantage of being thinner and smoother than the cast vane.

In the early development of turbines for high heads, considerable difficulty was experienced due to corrosion and wastage of the runner, even with water free from solid material in suspension. It is now generally accepted that such corrosion is due to faulty design of the runner vanes, leading to excessive eddy formation. The pressure at the core of such eddies is comparatively low, and this leads to the liberation from solution of air apparently containing nascent oxygen, which rapidly attacks and pits any iron or steel surface with which it may be in contact. Such corrosion can largely be eliminated by correct design. Where the supply water is clean, cast-iron runners are to be preferred as being smoother than cast-steel, except where the speed is so high as to necessitate cast steel being used to withstand the centrifugal stresses. Corrosion is pronounced at high heads if the turbines are operated much at part gate, while the practice of running for long periods without load is especially bad for the runners. Where the water carries grit or sand in suspension the runner is usually made of cast steel if large, and of phosphor bronze if small.

The general changes in the shape and proportions of the runner which have accompanied the recent development of the high-speed turbine are indicated diagrammatically in fig. 97, and in figs. 98† and 98*a*, which show one of the latest types of low-head turbine.‡ The change has been in the direction of increasing the depth of the buckets, and at the same time of maintaining or increasing the ratio of the discharge area at exit to that at entrance. Also, whereas it was formerly considered essential that the space between the guide vanes and the runner should be reduced to a minimum, the most recent low-head turbines show a very large radial clearance. In fact the latest type is, in its essentials, an axial-flow turbine, with the more convenient form of pivoted guide-vane regulation.

In a recent paper§ Nagler describes a new type of runner which is essentially similar to the impeller of a screw pump, and in which only from three to five vanes are used. This forms a purely axial-flow turbine. Development tests indicate that it is capable of extremely high rotative speeds under low heads, with reasonably high efficiencies.

End Thrust on Shaft. With a single-runner turbine, owing mainly to the static pressure behind the runner due to leakage between the runner

* By courtesy of the Cramp Shipbuilding Co., Philadelphia.

† By courtesy of Messrs. Piccard, Pictet, et Cie, Geneva, and of Messrs. Vickers, Ltd.

‡ See also Kaplan, *Zeitschrift fur des Gesamte Turbinenwesen*, 20th November, 1919.

§ *Am. Soc. Mech. E.*, December, 1919

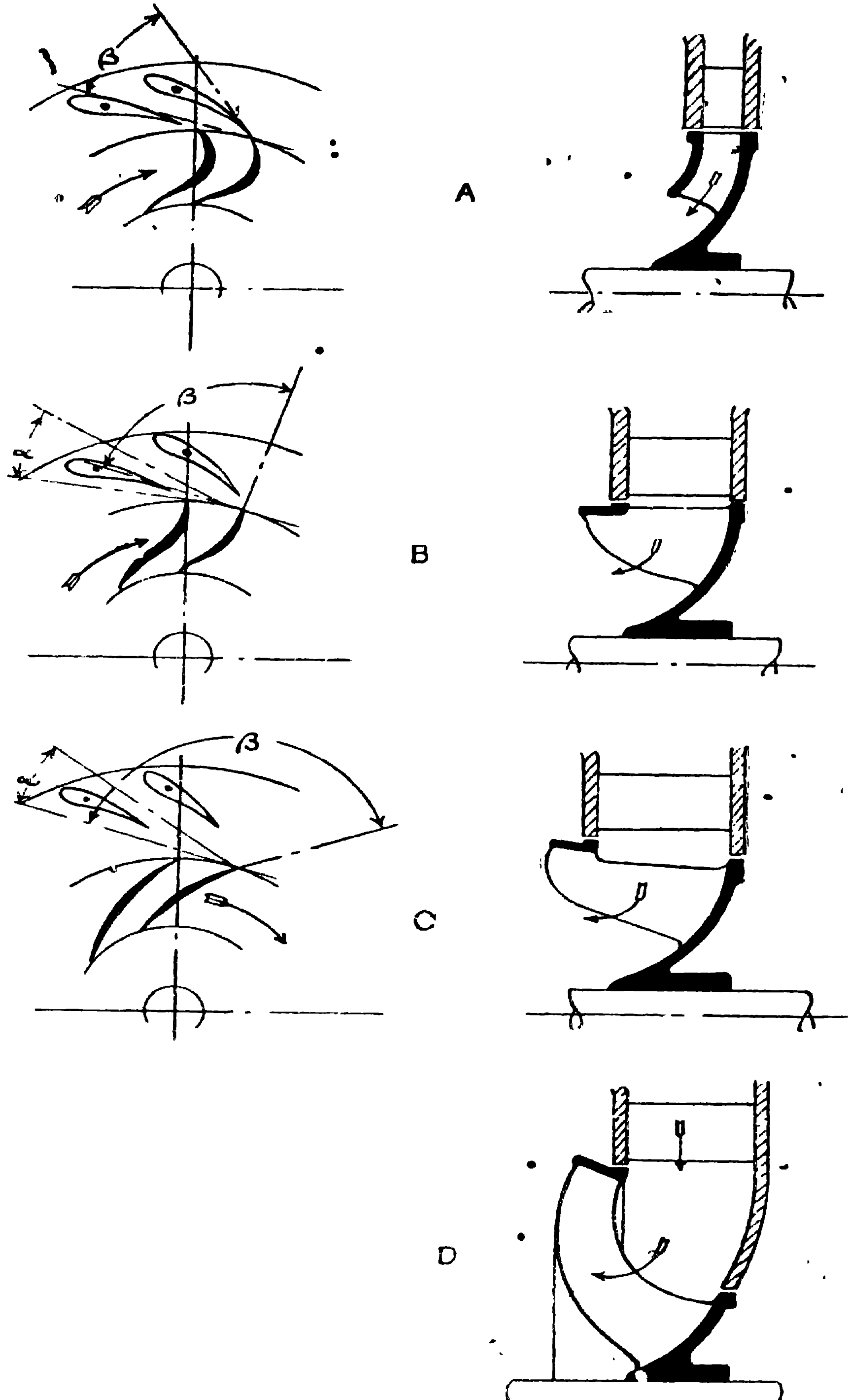


Fig. 97 — Types of Turbine Runner

and the casing, there is, unless special means are adopted to prevent it, an unbalanced end thrust on the shaft.

Where the head is low, the pressure behind the runner may be relieved by a series of vent holes through the runner crown. In order to prevent

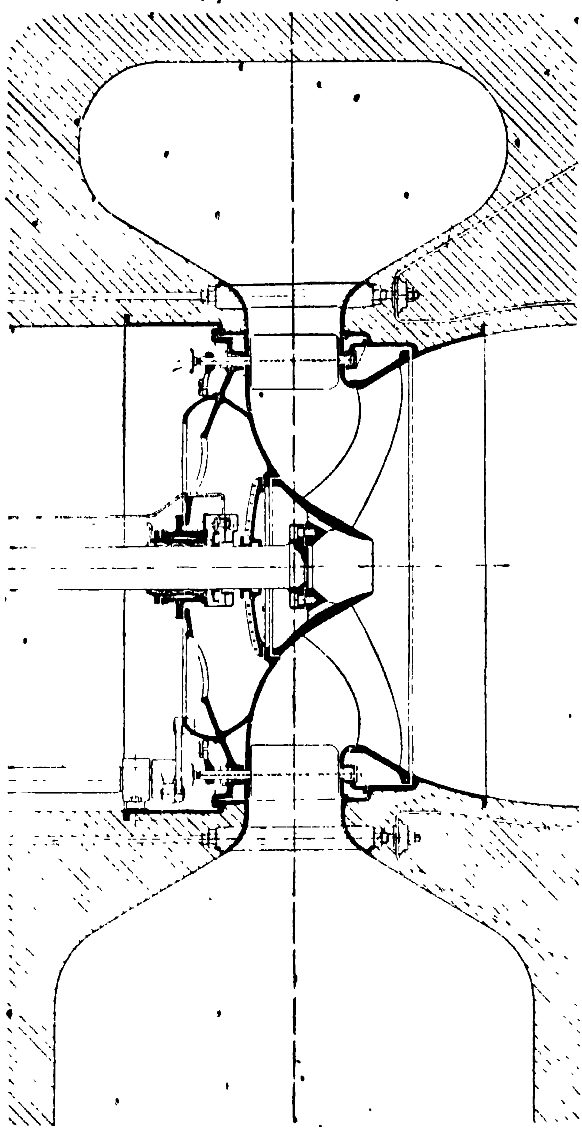


Fig. 98.—10,000 h.p. under 10 metres (32.8 ft.) head, at 93.7 r.p.m., specific speed 118.5. Efficiency at full load, 85.0 per cent.

the water behind the runner whirling, and so increasing in pressure outwards due to centrifugal action, a number of radial vanes, almost touching the wheel, are carried by the turbine casing. Any force still unbalanced is taken up by a small thrust block. Where the head is high this method is inadequate, and with a large turbine the end thrust is so large as to make

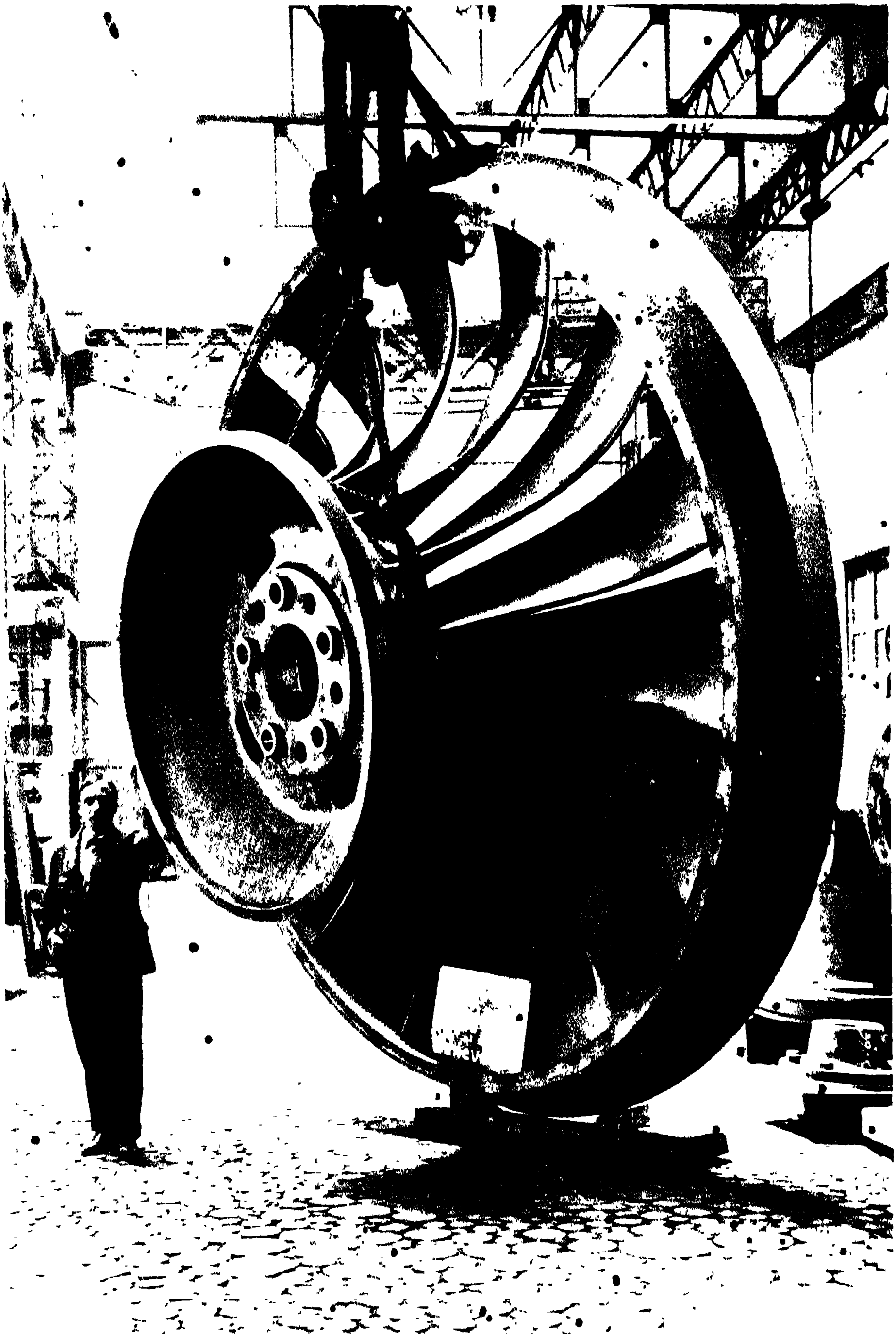


Fig. 66a. Runner for High-speed Turbine.

3400 b.h.p. at 7·2 ft. head. 88 per cent.

107 r.p.m.

the provision and maintenance of a suitable mechanical thrust bearing a matter of some difficulty. For this reason the greater part of the end thrust is balanced by hydraulic rather than mechanical means. One method of doing this is shown in fig. 94. The space to the right of the balancing piston is supplied with pressure water from the penstock through a regulating valve, while the space to the left is freely vented to the draft tube. Since there is a small leakage past the piston, the pressure in the balancing chamber may be closely regulated to suit the conditions of operation by adjustment of the regulating valve.

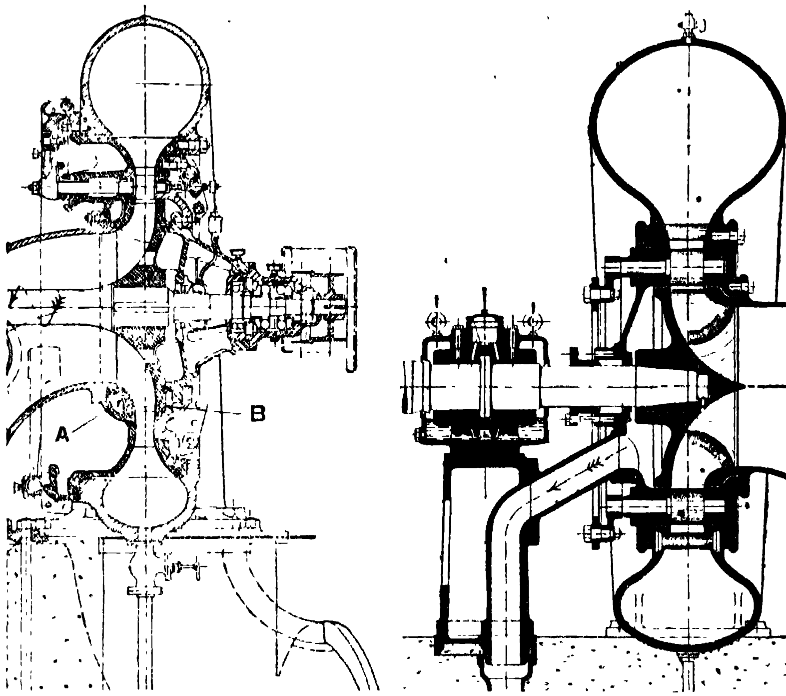


Fig. 99.—Balancing Devices for Turbine Runner

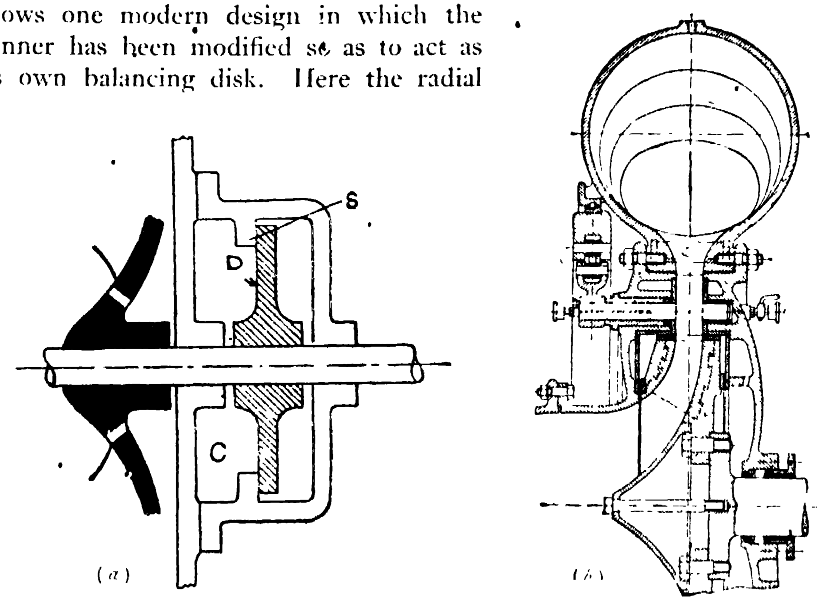
In a second method the areas of the runner at A and B (fig. 99 (a)) are made equal. Leakage from the spaces at A and B is reduced to a minimum by cutting down the radial clearance between runner and casing, and by the use of labyrinth joints. Any unbalanced force is taken up by a small thrust bearing.

In a third method, the space behind the runner is vented to the draft tube by a pipe of sufficiently ample dimensions to ensure the pressure in this space being sensibly the same as that in the draft tube (fig. 99 (b)).* Here again, any appreciable wear of the runner may render the discharge pipe inadequate in area, giving rise to an unbalanced force for which a thrust block must be provided.

Fig. 100 (a) shows a balancing arrangement in which no thrust block is

* Bergstrom. *Proc. Inst. Mech. E.*, Feb., 1920.

necessary. The space C to the left of the balancing disk D is supplied with pressure water from the penstock through a regulating valve, while the space to the right is freely vented. When the disk is in contact with the annular ring S, the force exerted by the pressure water is sufficient to overcome the force acting in the opposite direction on the runner, and the shaft moves to the right. A very small motion allows sufficient leakage between the disk and the seating ring to reduce the pressure in the chamber C to that necessary to produce equilibrium, and in practice the shaft oscillates laterally about this position within very narrow limits. In general the lateral movement does not exceed about ·01 inch. Fig. 100(*b*) * shows one modern design in which the runner has been modified so as to act as its own balancing disk. Here the radial clearance between the runner and the casing is reduced to a minimum. The two faced rings on the front and back faces of the runner work, with a normal clearance of about ·02 inch, over corresponding rings on the turbine casing. Any unbalanced force on the runner, say to the left, causes it to move to the right, increasing the clearance between the left-hand rings and reducing that between the right-hand rings. This reduces the pressure in the left-hand space, and increases that in the right-hand space until a position of equilibrium is attained.



(*a*) (*b*)

Fig. 100

Balance Piston for Vertical Turbines.—In the case of a vertical shaft turbine, installed at the bottom of a wheel pit and transmitting its power through a long vertical shaft, the vertical load to be carried is very large, and to obviate the necessity of carrying the whole of this load on a mechanical thrust bearing, balancing pistons have been used in a number of cases. Fig. 101 shows this method as applied in the case of the double

* Bergstrom, *Proc. Inst. Mech. E.*, February, 1920

Francis turbines of the Canadian Niagara Power Co. Here the weight of the rotating parts is 120 tons, and is balanced partly by the upward pressure on the bottom face of the lower runner, water under the full pressure of the supply head, 133 ft., being admitted to a balance chamber beneath this runner, and partly by the upward pressure on the rotating balance piston, which is subject to the same pressure. Leakage past this piston is drained away to the tail race, and by adjusting the valve on the supply pipe S the upward pressure may be regulated with great nicety.

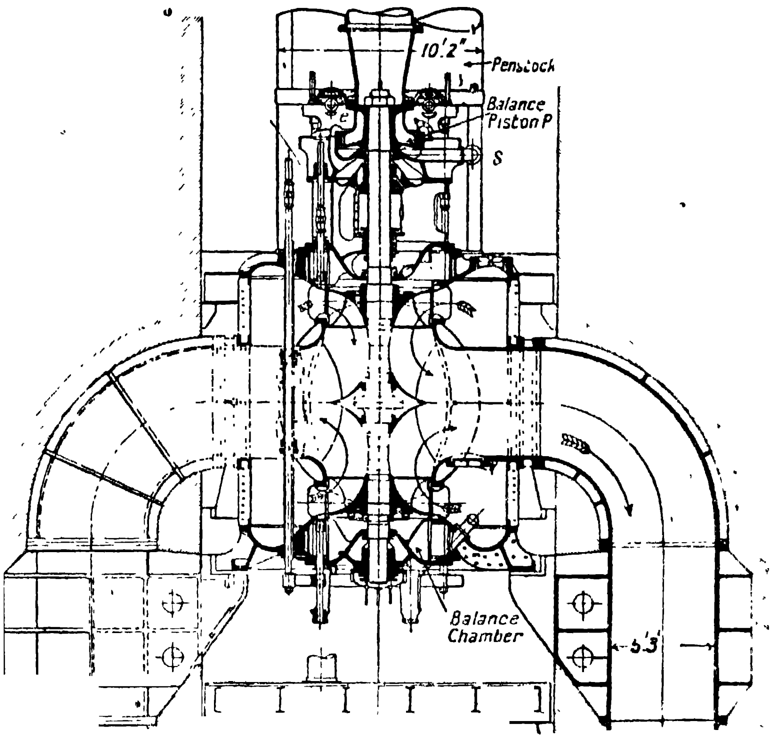


Fig. 101.—Balance Pistons for Vertical Shaft Turbine

Any unbalanced load is supported by the suspension bearing (fig. 102) which is placed on the upper deck. In this bearing, oil, under a pressure of 375 lb. per square inch, is supplied to the annular chamber C surrounding the bush B, and escapes outwards between the fixed and rotating disks at D. These disks have an outside diameter of 36 in. and a bearing area of 780 sq. in. Any slight swing or lateral wear of the shaft is permitted by the spherical bearing of the lower disk.

Thrust Bearings.—The thrust bearing for a horizontal shaft unit is usually of the simple collar type, incorporated as part of one of the main

bearings. Where the end thrust is comparatively small, ring oiled self-lubricating bearings or ball thrust bearings are satisfactory. For large pressures forced lubrication is advisable, with some cooling device for dissipating the heat generated at the bearing. The bearing may be water-jacketed, or the oil may be cooled by a cooling coil, containing circulating water, in

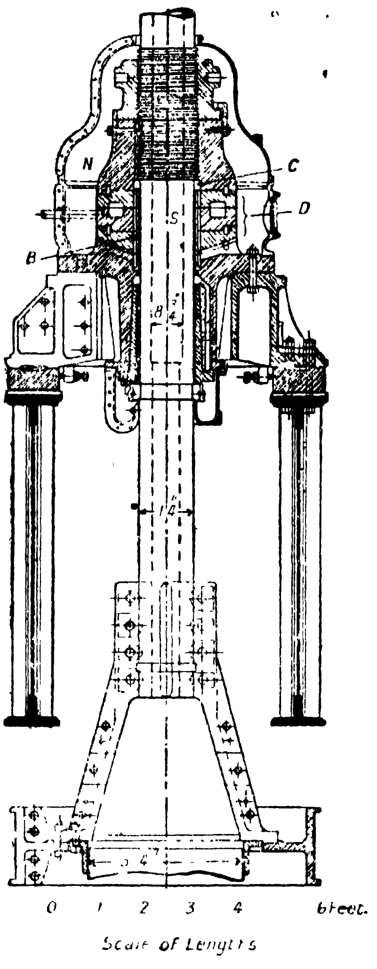


Fig. 102.—Suspension Bearing

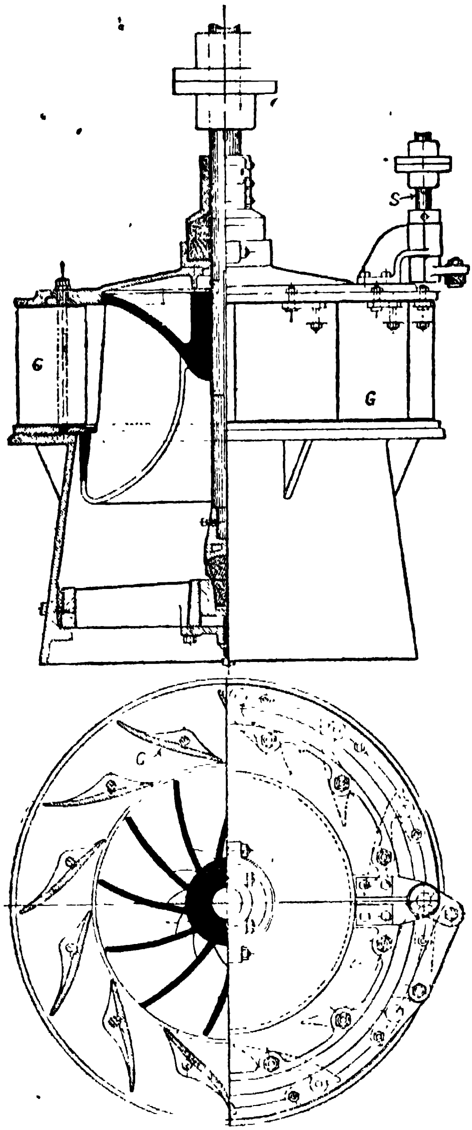


Fig. 103.—Vertical Shaft Turbine with Submerged Bearing

the oil tank. Bearing surfaces should be babbitted. A bearing forming part of or bolted to the turbine casing is preferable to one separately supported, since the latter is more likely to get out of alignment.

Bearings for Vertical Shafts.—In a small unit, erected in an open

forebay, the thrust bearing often consists of a plain footstep bearing supporting the lower end of the shaft (fig. 103), and provided with a lignum vitae bearing pad and bearing strips. In large units the bearing is never submerged. It may be carried by a cast-iron truss, supported from the speed ring of the turbine, and situated below the generator as shown in fig. 95, which illustrates one of the turbines of the Keokuk development on the Mississippi River, or may be situated above the generator on a supporting truss which forms the head cover of the generator, as shown in fig. 96, which shows a more recent development at the Cedars Rapids Power Co. The latter method has the advantage of being more accessible, and gives a more compact construction. The upper guide bearing of the generator is located immediately below the thrust bearing.

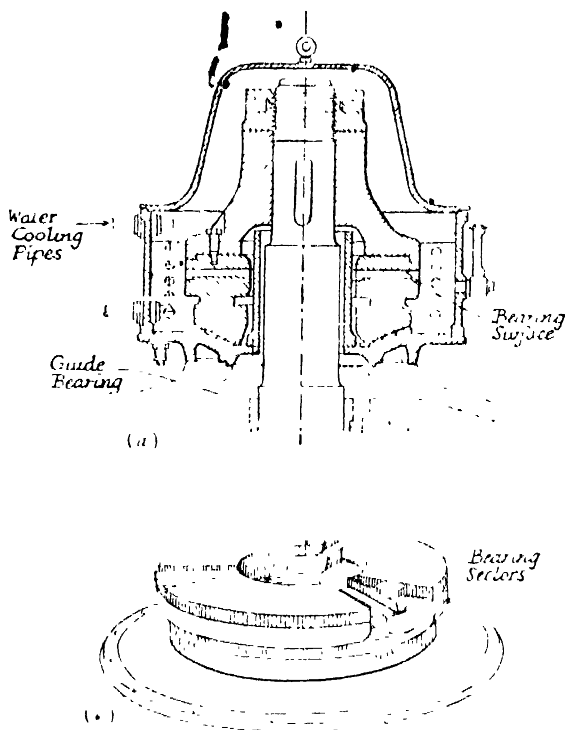


Fig. 104. Suspension bearing

Owing to the fact that the whole of the weight of the rotating parts of both turbine and generator is carried by the thrust bearing, this forms a very important feature in the design of such a unit. Until comparatively recently the type of oil-pressure bearing shown in fig. 102 was used almost exclusively for large units. The danger accompanying any failure of the oil-supply to such a bearing has led to the more general use, in modern units, of some form of bearing not requiring a pressure oil-supply. By replacing the lower disk by a series of blocks, each of which has an alternate inclined and flat sector, the oil, supplied under gravity, is fed into the segments by the rotation of the shaft, and a simple bearing (fig. 104 *a* and *b*) is produced which is capable of good results under large loads. Bearings of this type have been used for loads up to 80 tons* at speeds up to 375 r.p.m. Tests show them capable of withstanding pressures of approximately 1400 lb. per square inch at peripheral speeds of 70 ft. per second.

* At the Hauterive Installation (Fribourg, Switzerland).

An elaboration of this idea has led to the so-called Michell or Kingsbury bearing, in which the stationary disk is made up of several babbitted segments, each of which is mounted on a pivot to enable it to adjust its angle of inclination to the rotating disk, so as to ensure an approximately even distribution of loading over its surface. This bearing also has the advantage that the oil-supply does not need to be under pressure. In such bearings, applied to the Keokuk turbines (fig. 95), the diameter is 56 in., the total load 255 tons, and the mean bearing pressure 350 lb. per square inch. Tests show that the friction loss in these bearings amounts to between 7·5 and 10 kw. at the normal speed of 100 r.p.m., or approximately one-tenth of 1 per cent of the output of the turbine.

In some more recent installations a roller bearing of reduced dimensions is placed inside the main bearing. The roller bearing is made with a slight clearance, and only comes into operation if any wiping of the shoes of the main bearing causes the bearing plate to settle.

Roller bearings have given excellent results in many modern plants, under loads up to 250 tons and at speeds up to 300 r.p.m., while for light loads ball bearings are very satisfactory. Both types of bearing are sensibly independent of any oil-supply.

In some recent large units spring thrust bearings have been used in which a flexible plate divided into sections by from six to ten grooves takes the place of the pivoted blocks of the Michell bearing. This is supported by a large number of short, stiff spiral springs, with a view to obtaining an even distribution of loading over the bearing.

Guide Bearings. The guide bearing for a vertical shaft turbine may be either an oil-lubricated babbitted bearing or a lignum vitae bearing. The latter type has been fitted to a large number of modern units. It has the advantage of enabling the bearing to be brought closer to the runner than is feasible with the babbitted type, rendering the shaft less liable to develop vibrations. It also eliminates the necessity for an oil-supply and an oil-circulating pump in connection with the bearing, and thus removes one source of possible breakdown. The modern lignum vitae bearing consists of a large number of narrow longitudinal strips, dovetailed into the bearing boxes, with the end grain of the wood in contact with the shaft. Spaces for the circulation of water are allowed between adjacent strips. Clean, and, if necessary, filtered water is piped to the bearing, and this escapes through passages vented in the hub of the runner into the draft tube as shown in figs. 95 and 96. To prevent corrosion and to facilitate removal after wear has taken place, the shaft is bushed with bronze where it passes through the bearing and stuffing box.

87. The Draft Tube.—The suction or draft tube was originally designed with a view of enabling the turbine to be placed at a convenient height above tail-water level without loss of head. The maximum practical elevation depends on the diameter of the draft tube as indicated in the following table, which is adapted from values given by Meissner:

Diameter of tube, feet	1·0	1·5	2·0	2·5	3·0	4·0	5·0	6·0	8·0
Maximum elevation, feet	30·0	28·0	26·5	25·0	23·5	21·0	19·0	18·0	14·0

From these values should be subtracted the head equivalent to the velocity of flow down the tube.

The draft tube also, if well designed, enables a large proportion of the kinetic energy of discharge from the runner to be converted into pressure head, and so to be utilized. The mean velocity of discharge from the runner varies from about $\cdot 2\sqrt{2gH}$ in a slow-speed turbine under high head to $\cdot 5\sqrt{2gH}$, or even more, in a high-speed turbine under a low head. Adopting the latter value, the kinetic energy of discharge represents 25 per cent of the total head, and for high efficiencies in such an installation it is essential that a large proportion of this energy should be recovered. In fact, the high efficiencies of modern low-head plants have only been rendered possible by the most careful attention to this part of the installation.

With a parallel draft tube discharging vertically into the tail race, the whole of the kinetic energy is lost. If, however, the tube is designed with a gradually increasing diameter, so that the velocity is gradually reduced from v_1 to v_2 before discharge, the loss of energy, which depends on the angle θ, included between the opposite sides of the tube, is reduced. Experiments on comparatively small tubes * show that this loss, expressed as a fraction of $(v_1 - v_2)^2 \div 2g$, the loss obtaining with a sudden change of velocity from v_1 to v_2, is as indicated in the following table:

Conical angle θ		5°	10°	15°	20°	25°
Loss of head in tube expressed as a fraction of $\frac{(v_1 - v_2)^2}{2g}$		·13	·17	·27	·42	·62
Total loss of head expressed as a fraction of $\frac{v_1^2}{2g}$, where	$v_2 = \frac{v_1}{4}$ ..	·13	·16	·21	·30	·41
	$v_2 = \frac{v_1}{3}$ ..	·17	·19	·23	·30	·39
	$v_2 = \frac{v_1}{2}$ ..	·28	·29	·32	·35	·40

The last three lines of this table show the total loss of head, including that due to the final velocity v_2, when v_2 is respectively one-fourth, one-third, and one-half v_1.

With a given length of draft tube, the choice lies between a small

* *Trans. Roy. Soc. Edinburgh*, Vol. 48, 1911, p. 97. Also *Hydraulics*, Gibson (Constable & Co., 1912), pp. 85–92.

angle of flare with a relatively small ratio of v_1 to v_2, or a larger angle and a larger velocity ratio. The latter gives a bigger loss in the tube but a smaller loss at discharge, and the best proportions are such as give the minimum total loss. If, for example, the diameter of the runner at exit is 4 ft., and the available length of draft tube is 20 ft., an angle of flare of 5° will make the outlet diameter 5·75 ft., the ratio of inlet and outlet velocities 2·06, and the total loss of head $\cdot 27 v_1^2 \div 2g$. An angle of 10° will make the outlet diameter 7·5 ft., the ratio of velocities 3·52, and the loss of head $\cdot 18 v_1^2 \div 2g$, while an angle of 15° will make the outlet diameter 9·27 ft.,

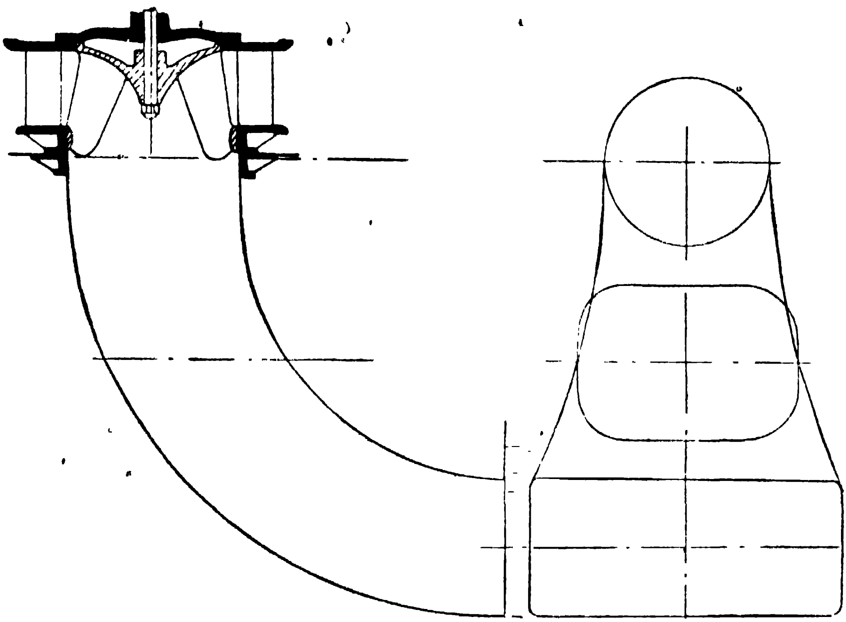


Fig. 105. Draft Tube.

the ratio of velocities 5·37, and the loss of head $\cdot 21 v_1^2 \div 2g$. In this case the best angle of divergence would be slightly over 10°.

The diameter of the draft tube at entrance should be the same as that of the runner to avoid all shock at this point. By curving the tube so as to discharge in the direction of flow in the tail race, the kinetic energy of its discharge is partially utilized in assisting this flow, and is not entirely wasted.

The draft tube may be either of steel-plate or cast-iron construction, or may be moulded in the concrete substructure, as shown in fig. 146 148, p. 211. The latter arrangement, if not ruled out on the ground of expense, is preferable where circumstances permit, since it enables any desirable form of cross section to be adopted, whereas with a steel-plate construction the conical circular form is almost essential. With a moulded draft tube, a form such as fig. 105, which enables a comparatively shallow tail race to be used, presents no difficulties.

With a horizontal shaft unit, the turbine shaft passes through the draft tube, and a stuffing box is necessary to prevent air leakage into the tube. Air tightness may be assured by means of a water seal consisting of a chamber C surrounding the shaft (fig. 106), and supplied with pressure water from the penstock through a small pipe P.

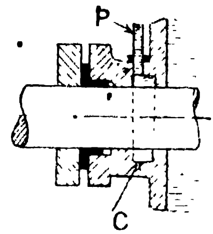


Fig. 106

Incidentally a conical draft tube tends to improve the speed regulation. With quick regulation on a falling load the inertia of the suction column tends to break the column and to produce a vacuum in the turbine casing. Any such separation is followed by a reflux up the draft tube, which may give rise to severe water-hammer effects. Even a comparatively small change of load may, with a long tube, set up such pulsations, which are detrimental to steady running, and which are reduced by the use of a conical draft tube. As an example of this effect, consider the case of a vertical parallel draft tube l ft. long, dipping h_d ft. below the surface in the

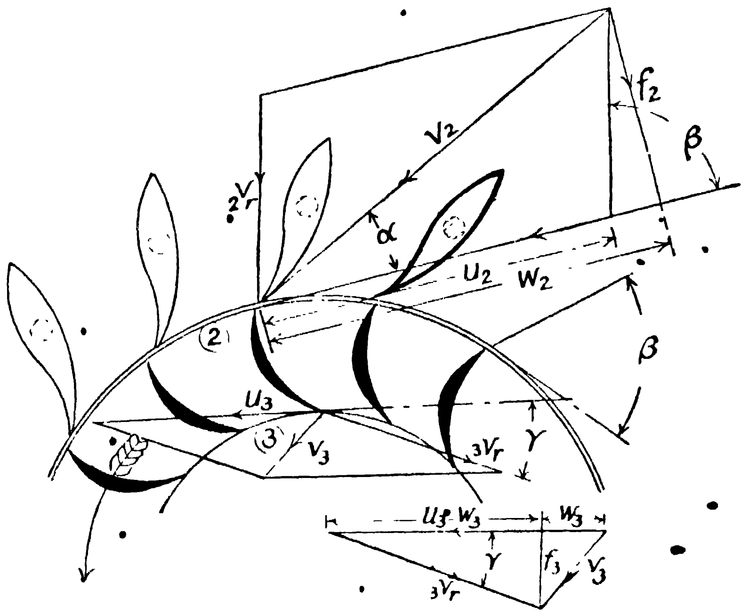


Fig. 107

tail race, and suppose air leakage at the top of the draft tube to increase the pressure by the equivalent of h_a ft. of water. If then a be the retardation, and if the barometric height is 33 ft., for separation to occur

$$33 < (l - h_d) - h_a + \frac{l}{g}a.$$

$$a = \frac{g}{l}(33 - l + h_d - h_a).$$

If, for example, $l = 28$, $h_d = 2$, $h_a = 4$, the necessary retardation is only 3·43 ft. per second per second.

88. Hydraulics of the Reaction Turbine.—The space available renders it impossible to do more than touch on the general principles on which the hydraulic design of the reaction turbine is based.*

In the following discussion let

w' = angular velocity of the runner in radians per second,
$w = 2\pi N \div 60$ where N = revolutions per minute,
$u = wr$ = velocity of wheel at point indicated by a suffix,
v = absolute velocity of water,
w = tangential component of v,
f = radial component of v,
v_r = relative velocity of water and vane,
α = guide vane angle (fig. 107),
β = wheel vane angle at entrance,
γ = wheel vane angle at exit,
Q = flow in c.f.s.,
W = weight of 1 c. ft. of water,
suffix (2) refers to inlet to wheel vanes,
,, (3) refers to exit from wheel vanes

For entry without shock, the direction of the relative velocity of water and vane at entrance to the wheel is to be parallel to the vane tips, and a consideration of the diagram of velocities (fig. 107) shows that if the angles are correctly proportioned:

$$f_2 = w_2 \tan \alpha = (w_2 - u_2) \tan \beta; \quad \therefore u_2 = w_2\left(1 - \frac{\tan \alpha}{\tan \beta}\right).$$

$$f_3 = (u_3 - w_3) \tan \gamma.$$

$$_2v_r = f_2 \operatorname{cosec} \beta; \quad _3v_r = f_3 \operatorname{cosec} \gamma.$$

$$\left.\begin{array}{l}\text{The change of the moment of momentum}\\ \text{in the wheel = turning moment}\end{array}\right\} = \frac{WQ}{g}\{w_2r_2 - w_3r_3\} \text{ ft. lb.}$$

$$\left.\begin{array}{l}\therefore \text{Work done per second}\\ \text{on runner}\end{array}\right\} = \frac{WQ}{g}\{w_2r_2 - w_3r_3\} w$$

$$= \frac{WQ}{g}\{w_2u_2 - w_3u_3\} \text{ ft. lb.}$$

In an ideal wheel, with no friction or eddy losses, we should have, neglecting changes of level in the wheel,

$$\frac{p_2}{W} + \frac{v_2^2}{2g} = \frac{p_3}{W} + \frac{v_3^2}{2g} + \text{work done per pound between (2) and (3).}$$

The efficiency will be a maximum when the energy rejected in the discharge is a minimum, i.e. when v_3 is a minimum, or when w_3 is zero, in which case

* For further information the reader is referred to some such book as that of Gelpke and van Cleeve, *Turbines and Turbine Installations*, or *Vorlesungen uber Wasserkraft Maschinen*, by Dr. R. Camerer.

$v_3 = f_3$. Assuming the wheel to be designed for this state of affairs,

$$\frac{p_2}{W} + \frac{v_2^2}{2g} = \frac{p_3}{W} + \frac{f_3^2}{2g} + \frac{w_2u_2}{g}.$$

Writing H as the head available to produce flow through the wheel, so that

$$H = \frac{p_2}{W} + \frac{v_2^2}{2g} - \frac{p_3}{W}, \text{ we have } H = \frac{f_3^2}{2g} + \frac{w_2u_2}{g},$$

from which, writing $f_3 = f_2\dfrac{b_2r_2}{b_3r_3} = w_2 \tan\alpha \dfrac{b_2r_2}{b_3r_3}$,

where b and r are the breadth and radius of the wheel, we get, on substitution,

$$w_2 = \sqrt{\frac{2gH}{2 + \left(\frac{b_2r_2}{b_3r_3}\tan\alpha\right)^2 - 2\frac{\tan\alpha}{\tan\beta}}}$$

$$\text{while } u_2 = w_2\left(1 - \frac{\tan\alpha}{\tan\beta}\right),$$

$$\text{and } Q = A_2f_2 = A_2w_2\tan\alpha.$$

Thus in a wheel of given design, the peripheral speed for maximum efficiency and the volume of discharge each vary as $\sqrt{H}$, while the output of the turbine, being proportional to Qu_2w_2, varies as $H^{\frac{3}{2}}$.

$$\text{The hydraulic efficiency } \eta = \frac{\text{work done per pound}}{H}$$

$$= \frac{w_2u_2}{gH},$$

which on substitution becomes $\dfrac{1}{1 + \frac{1}{2}\left(\frac{b_2r_2}{b_3r_3}\tan\alpha\right)^2\left(\frac{\tan\beta}{\tan\beta - \tan\alpha}\right)}$.

Writing $u_2 = k\sqrt{2gH}$, the manner in which k, and therefore the speed of the wheel, and in which η vary with changes in α and β is shown in the following table for the special case in which $b_2r_2 = b_3r_3$.

Values of β.		Values of α.				
		10°	15°	20°	25°	30°
60°	k	·658	·636	·604	·564	·516
	η	·981	·959	·922	·871	·800
90°	k	·702	·695	·685	·672	·655
	η	·984	·964	·936	·902	·857
120°	k	·741	·748	·756	·764	·770
	η	·986	·968	·946	·920	·889
150°	k	·802	·845	·885	·924	·960
	η	·988	·971	·959	·940	·917

The results show that by suitable adjustment of α and β, the peripheral speed for a given head may be varied between wide limits. For high speeds β should be large, and for high efficiencies α should be small. As β is increased, the value of f_2 and hence the volume of water passing a wheel of given size diminishes, so that to obtain the same output the size of the wheel is to be increased. If, as is usually the case in low-head plants, a high rotative speed is required, the inlet area is increased by increasing the depth of the runner. Such a turbine has a comparatively large ratio of inlet area to discharge area, and the velocities of discharge are relatively high. For high heads β may be between 60° and 90°, and, for medium and low heads, between 90° and 135°.

Similarly, while the hydraulic efficiency decreases as α increases, the volume of flow increases with α, and the maximum output is obtained when the product of Q and η is a maximum. For high efficiency α should be as small as mechanical considerations permit, generally between 12° and 18°. Where a cheap turbine is required, and the efficiency is not of great importance, α may have a value as high as 35°.

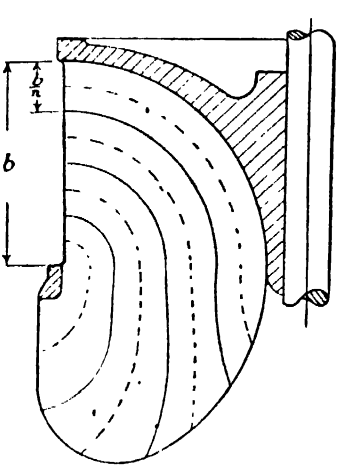


Fig. 108

The foregoing theory neglects the effect of losses of head due to eddy formation at the entrance to and in the runner, of leakage between the runner and its casing, and of mechanical friction, and assumes that all the water filaments have the same directions of motion at entrance and exit, and that these directions are parallel to the directions of the vane tips at these points. In any commercial turbine this latter assumption is not nearly true, and in such a case the runner is imagined to be divided into a series of elements (fig. 108) in each of which this condition is approximately fulfilled, and each of which has its own appropriate vane angles. These elements are then combined to give the complete runner. The effect of the various hydraulic losses can be taken into account by the introduction of suitable constants based on experimental tests on similar runners, but the true value of the theory lies mainly in its power of indicating the relative influence of the main points of design on the efficiency and output, and in its ability to give a preliminary design which may afterwards be modified in detail by the results of tests.

89. Model Tests. The more complex the form of the runner, the more necessary does it become to rely on experiment for the final design, and for an estimate of the performance and output under varying conditions of speed, gate opening, and head, and the very high efficiencies attained by some modern turbines have only been obtained by correct theoretical design aided by extended experimental investigation. Testing plants are

now utilized by most high-class manufacturers, in which all standard runners are accurately analysed, and in which new designs are tested. In the case of a large turbine, a geometrically similar model is tested. For such tests it is not necessary that the head should be the same as in the large turbine. If the suffix m refer to the model, and if S be the scale ratio of runner and model, and N the revolutions per minute, then if the turbine and its model are installed in similar settings, their efficiencies will be very approximately equal if

$$\frac{N_m}{N} = S\sqrt{\frac{H_m}{H}}$$

and if, at the same time, the volumes of water passing are in the ratio

$$\frac{Q_m}{Q} = \frac{1}{S^2}\sqrt{\frac{H_m}{H}},$$

the horse-powers will then be in the ratio

$$\frac{\text{H.P.}}{(\text{H.P.})_m} = S^2\left(\frac{H}{H_m}\right)^{\frac{3}{2}}.$$

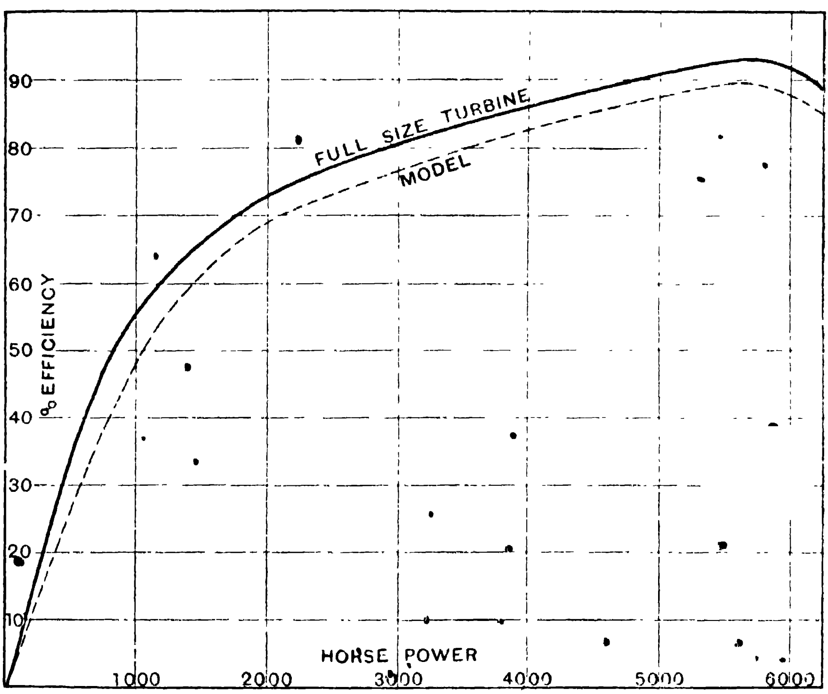


Fig. 109.—Efficiency Curves from Francis Turbine developing 6000 h.p. at 116 r.p.m., under a head of 49 ft.

The relative effect of friction is somewhat greater in the model than in the full-size machine, and as the losses in the setting are usually also greater

in the model, the efficiencies obtained from tests on the complete installation are often rather higher than from the model tests.

Fig. 109 shows such efficiency curves as obtained from a Francis turbine developing 6000 h.p. at 116 r.p.m. under a head of 40 ft.* The runner diameter is 7 ft. $6\frac{1}{4}$ in., while that of the model is 2 ft. $3\frac{3}{8}$ in. In

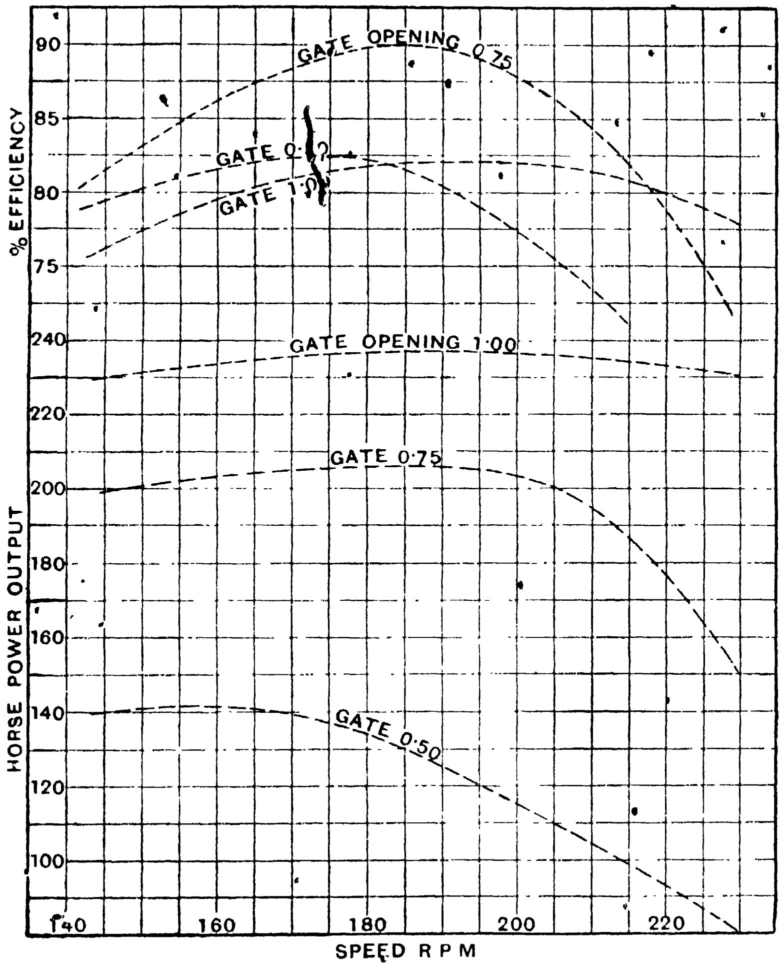


Fig. 110.—Speed-efficiency and Speed-power Curves

this case the design of the setting in the large plant was much better than in the model tests. Where the settings are similar, the coincidence between the two curves is usually closer than in this particular example.

90. Characteristic Curves for Reaction Turbines.—During the tests of a turbine or model, the head is usually maintained as nearly as possible constant, and the output and efficiency are measured for a series of gate openings and speeds. From these results, speed-power and speed-

* H. B. Taylor, *Gen. El. Review*, June, 1914, Appalachian Power Co.

efficiency curves as shown in fig. 110 may be drawn for a series of gate openings. In order, however, to show graphically the behaviour of the turbine under a wide range of operating conditions, a series of "characteristic curves" is usually prepared from the test data. Such a diagram, deduced from tests on a Francis turbine designed for 124 b.h.p. at a speed of 187 r.p.m. under a head of 17·3 ft., is shown in fig. 111*. To construct this diagram, the values obtained from the tests and as shown in fig. 110 are reduced to those corresponding to a unit head of 1 ft. The speed-power

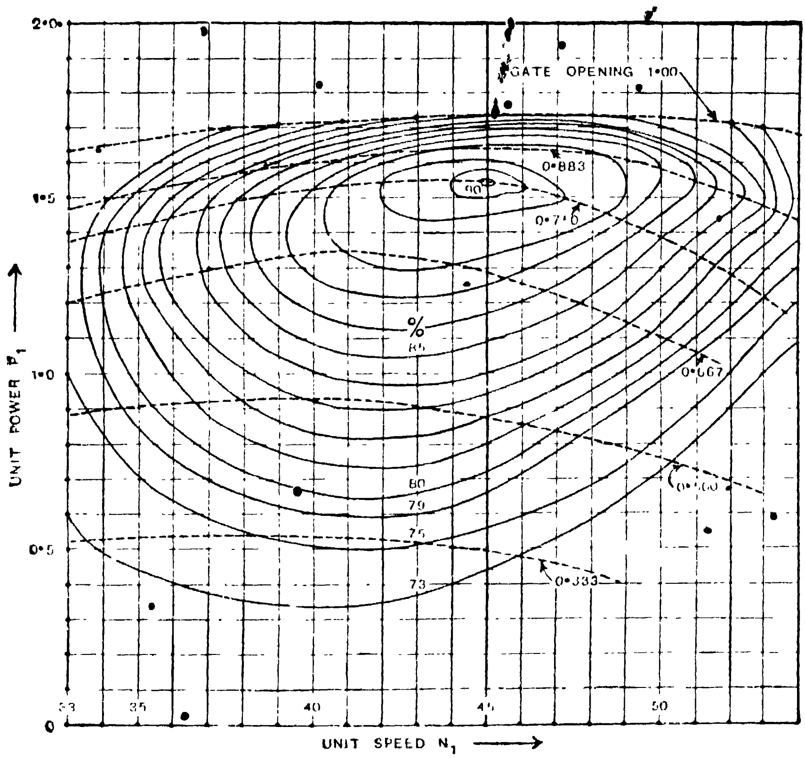


Fig. 111.—Characteristic Curves for Francis Turbine having a specific speed of 59

curves under this head at the various gate openings are then drawn as shown in the dotted lines of figs. 111 and 112 on a base of unit speeds, the unit speed being $N \div \sqrt{H}$. In the above case the unit speed is $187 \div \sqrt{17 \cdot 3} = 45 \cdot 0$ r.p.m. From the speed-efficiency curves, the speeds at which the same efficiencies are obtained under different gate openings are obtained. These speeds are marked on the corresponding speed-power curves, and through these points a series of continuous curves are drawn. These are the "characteristic curves".

This type of diagram is valuable in an investigation of the effect of any probable change in operating conditions, or of the suitability of the turbine

* Bergstrom, *Proc. Inst. M. E.*, February, 1920.

for conditions other than those for which it is designed. If, for example, this particular turbine were required to operate under a 20-ft. head, the power developed at full gate would be

$$1{\cdot}73 \times 20^{\frac{3}{2}} = 154{\cdot}8 \text{ b.h.p.,}$$

and the normal speed would be

$$45 \times \sqrt{20} = 201 \text{ r.p.m.,}$$

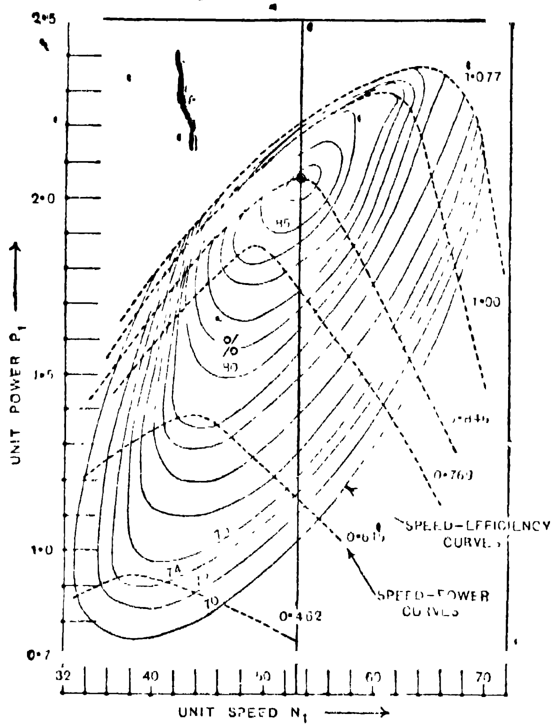


Fig. 112.—Characteristic Curves for Francis Turbine having a specific speed of 78·5

while the efficiencies under these conditions would be the same as shown in the diagram for the normal unit speed. If, however, the speed were required to be say 220 r.p.m., the corresponding unit speed would be

$$220 \div \sqrt{20} = 49{\cdot}2 \text{ r.p.m.}$$

The corresponding efficiencies may be read off the diagram. For this turbine the values are shown in the second line of the following table. The efficiencies at normal speed are given in the third line of the table.

Gate opening	1·0	·883	·750	·667	·500
Efficiency, per cent. . .	82·8	87·6	86·4	83·0	75·5
Efficiency at normal speed . .	83·0	88·2	90·0	87·3	82·0

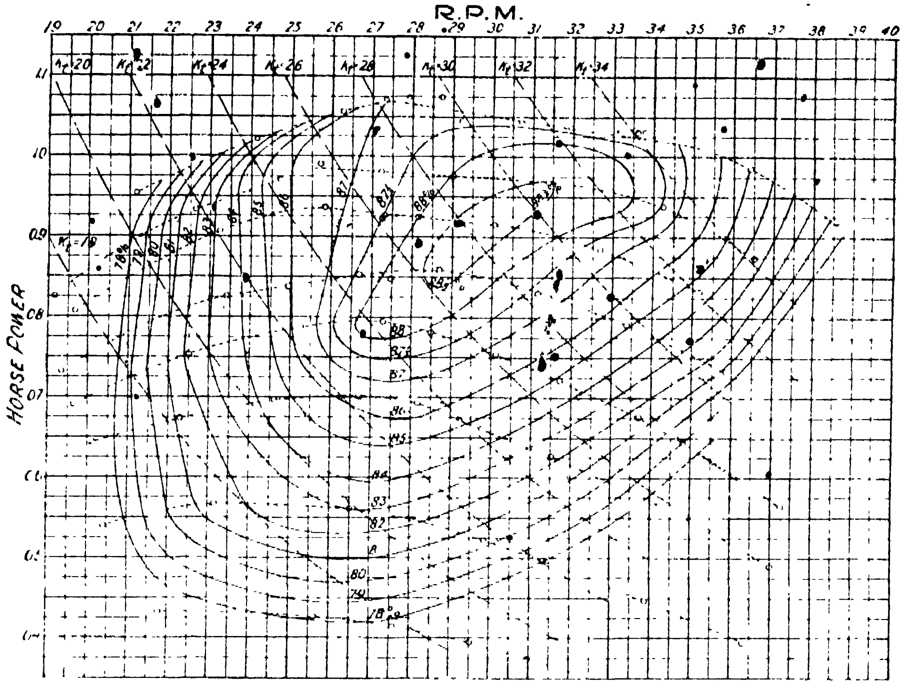


Fig. 113.—Characteristic Curves of 44-in. Francis Turbine, specific speed range from 18 to 34

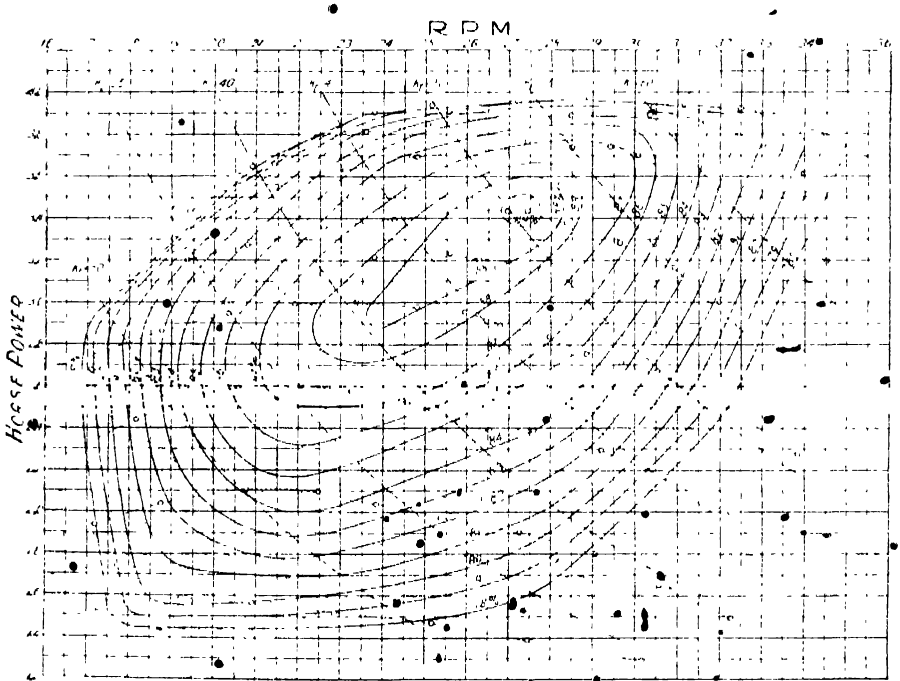


Fig. 114.—Characteristic Curves of 53-in. turbine specific speed range from 30 to 60

If the speed of 220 r.p.m. were to be maintained, and the head to be increased to say 30·3 ft., the unit speed would become

$$220 \div \sqrt{30{\cdot}3} = 40{\cdot}0 \text{ r.p.m.}$$

The unit power at full gate is now 1·71, and the output is

$$1{\cdot}71 \times 30{\cdot}3^{\frac{3}{2}} = 285 \text{ b.h.p.}$$

The efficiency under these conditions would be as follows:

Gate opening ..	1·0	·883	·750	·667	·550
Efficiency per cent ..	80·6	85·4	86·0	86·5	83·0

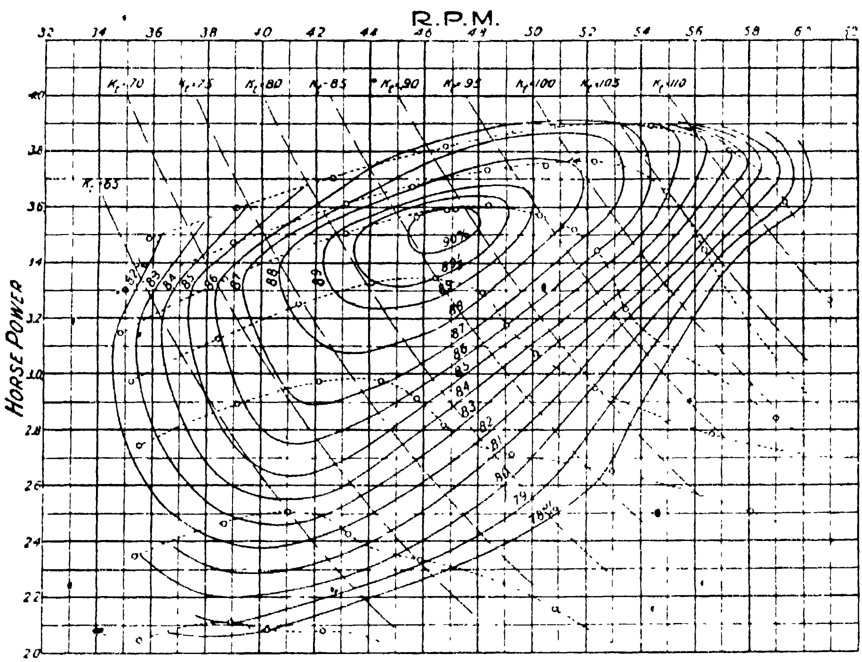


Fig. 115.—Characteristic Curves of 30-in. turbine, specific speed range from 65 to 110.

This particular turbine has a specific speed of 59. Fig. 112* shows the characteristic curves for a turbine having a normal specific speed of 78·5.

Figs. 113–15 † show characteristic curves of a series of turbines having specific speeds ranging from 18 to 110.

91. Specific Speed of a Turbine.—If P denote the output in b.h.p., then, for a given turbine,

* *Proc. Inst. Mech. E.*, February, 1920.
† By courtesy of Messrs. S. Morgan Smith Co., York, Pa.

(1) the speed N is proportional to $\sqrt{H}$,
(2) the discharge Q is proportional to $\sqrt{H}$,
(3) the output P is proportional to $H\sqrt{H} = H^{\frac{3}{2}}$.

In order to afford a basis of comparison of turbines of different diameters and proportions operating under different heads, the term known as "specific speed" has been introduced. This may be defined as the speed at which a runner would operate if reduced geometrically to such a size that it would develop 1 h.p. under unit working head. The figures for specific speed given in the following pages refer to a unit head of 1 ft. If the metre be adopted as the unit, these figures require to be multiplied by 4·45.

To determine the value of the specific speed, imagine the head to be reduced from H to h, the dimensions remaining unaltered. Then since the peripheral speed for maximum efficiency is proportional to $\sqrt{H}$, while the output is proportional to H, we have:

$$\frac{n}{N} = \sqrt{\frac{h}{H}}, \text{ while } \frac{p}{P} = \left(\frac{h}{H}\right)^{\frac{3}{2}}.$$

Now imagine all the proportions of the turbine to be reduced in the same ratio. Since for a given head the number of revolutions is proportional to the diameter, and since the quantity of water is proportional to the inlet area, and therefore to the square of the diameter, we have:

$$\frac{p}{P} = \left(\frac{h}{H}\right)^{\frac{3}{2}} \times \left(\frac{d}{D}\right)^{2}$$

$$\text{and } \frac{n}{N} = \sqrt{\frac{h}{H}} \times \frac{D}{d}$$

$$= \sqrt{\frac{h}{H}}\sqrt{\frac{P}{p}} \cdot \left(\frac{h}{H}\right)^{\frac{3}{4}}$$

$$= \sqrt{\frac{P}{p}} \cdot \left(\frac{h}{H}\right)^{\frac{5}{4}}.$$

If now h and p be made equal to unity, n becomes the specific speed N_s, so that

$$N_s = \frac{N\sqrt{P}}{H^{\frac{5}{4}}}.$$

In the case of a turbine having more than one runner, P in this expression represents the output per runner. If P is taken as representing the total output, this value of N_s is to be divided by the square root of the number of runners.

The specific speed of a reaction turbine may be varied by varying the diameter of the runner, the angle of the guide vanes, and the angle of the wheel vanes. By modifying the design as indicated in the sketches of fig. 97 it is possible, while maintaining high efficiencies at full load, to

increase the specific speed from about 15, its minimum value with the type shown in fig. 97A, to about 125 with the type shown in fig. 97D. Specific speeds as high as 150 are possible with some sacrifice in efficiency, and it is probable that further developments will see the value increased still further. The turbine shown in fig. 98 has a specific speed of 118·5 at its designed speed of 93·7 r.p.m. under a head of 32·8 ft. Under these conditions efficiency at full load is 85·0 per cent, at three-quarters load 81 per cent, and at half load 71 per cent. The corresponding discharges are 3110, 2470, and 1910 c. ft. per second. Under a head of 19·7 ft. at the same speed of rotation its output is 4200 h.p., its specific speed 146, and its full load efficiency 68 per cent. The type of runner shown in fig. 97A is well adapted for high heads and relatively small volumes; B is suitable for medium speeds; C for high speeds and medium heads; and D for low heads and high speeds. These high specific speeds are extremely valuable for low-head installations, since they enable the size and cost of the turbine, of its setting, and of the generator to be greatly reduced. In fact, many existing low-head installations would have been commercially impracticable but for the development during recent years of the high-speed turbine.

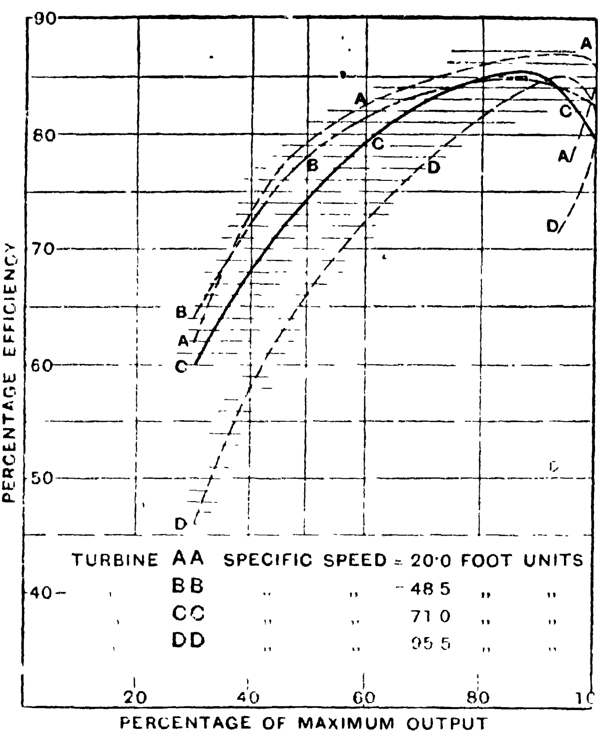


Fig. 116. Variation of Efficiency with Output

High specific speeds are, however, attended by some disadvantages, particularly for medium and high heads. The part gate efficiency in general falls off as the specific speed increases, broadly as indicated by the curves of fig. 116, which show the efficiencies of a series of model test turbines of approximately the same size operating under the same head of 4 m.* Also if the speed is unduly high it becomes very difficult to avoid corrosion troubles. At the present stage of design, the maximum specific speeds to be used under normal circumstances with various heads are

* By the courtesy of Messrs. Boving & Co., Ltd.

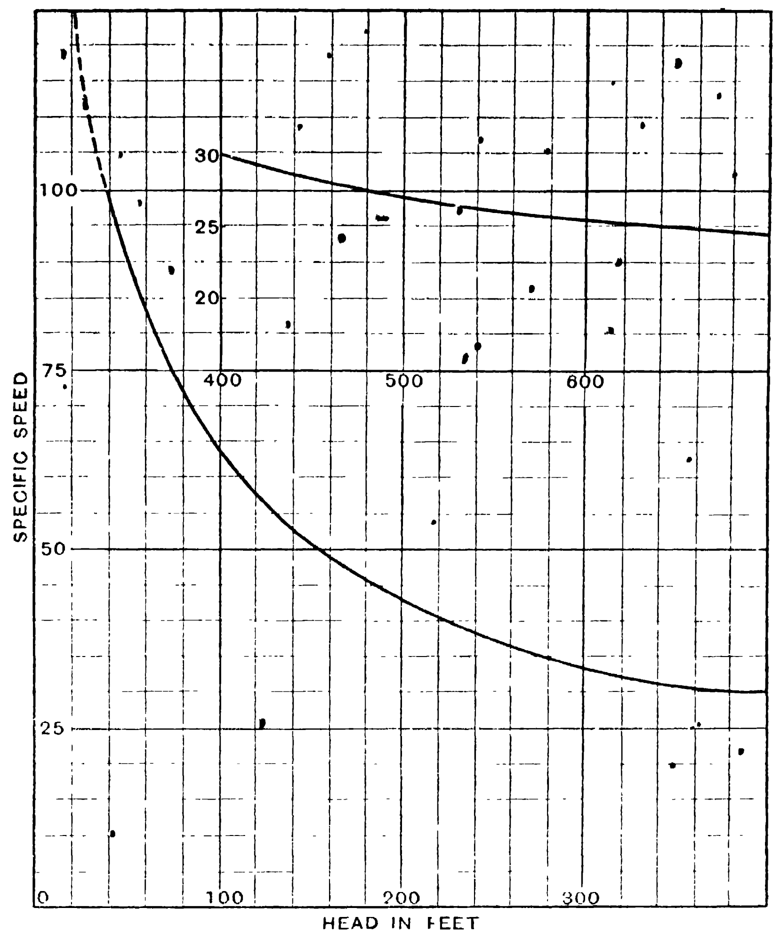


Fig. 117

approximately as represented by the curves of fig. 117.

In the earlier low-head plants, sufficiently high speeds were attained by the use of two or more runners on the same shaft as shown in fig. 145. (See pocket at end of volume.) This arrangement has a number of disadvantages as compared with the single runner. These are:

1. A separate series of guide vanes or gate mechanisms is required for each runner, one or all of which are submerged.

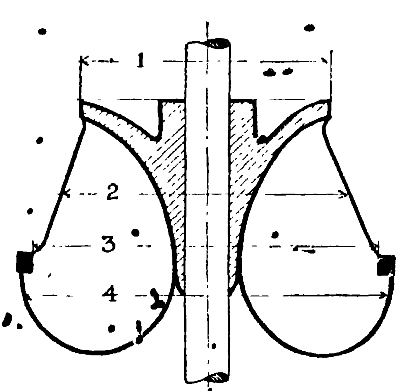
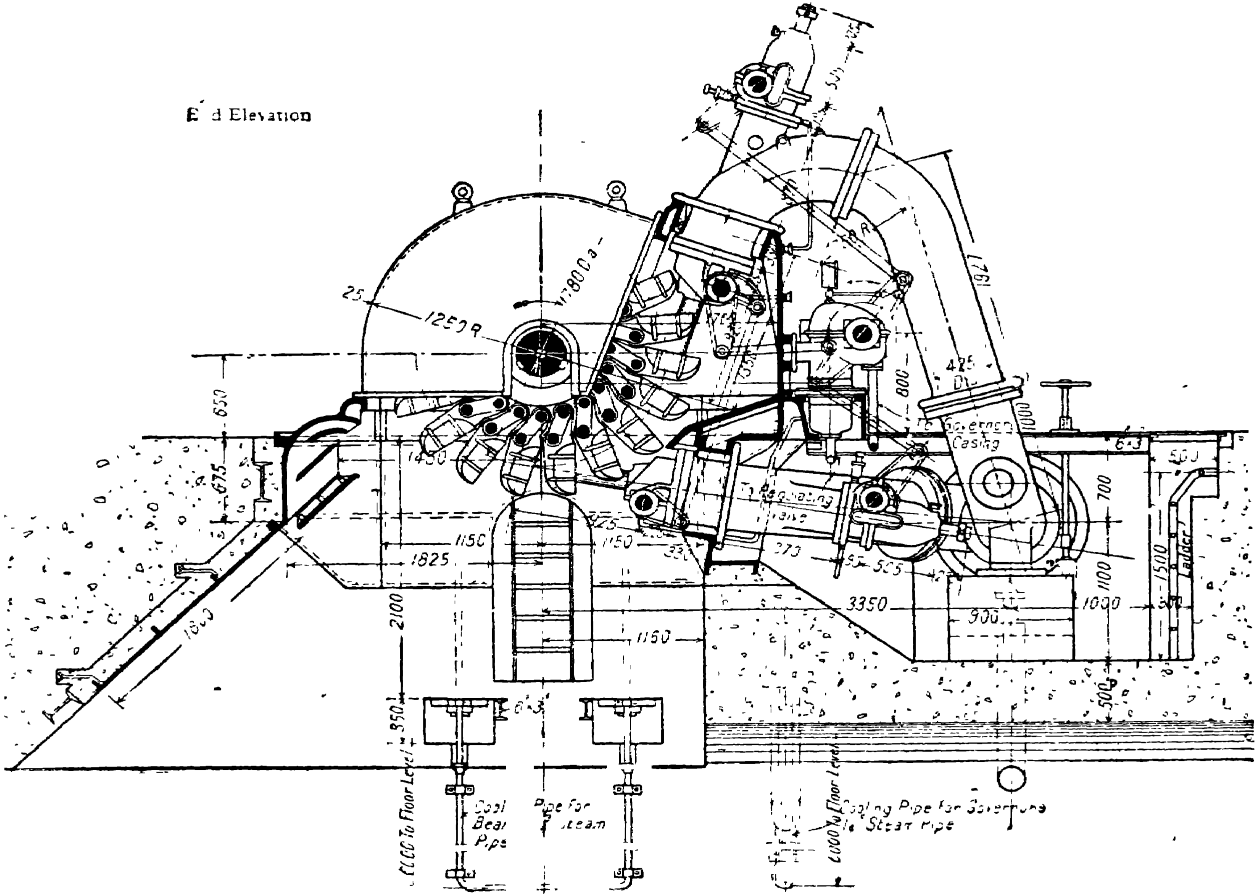


Fig. 118

E d Elevation



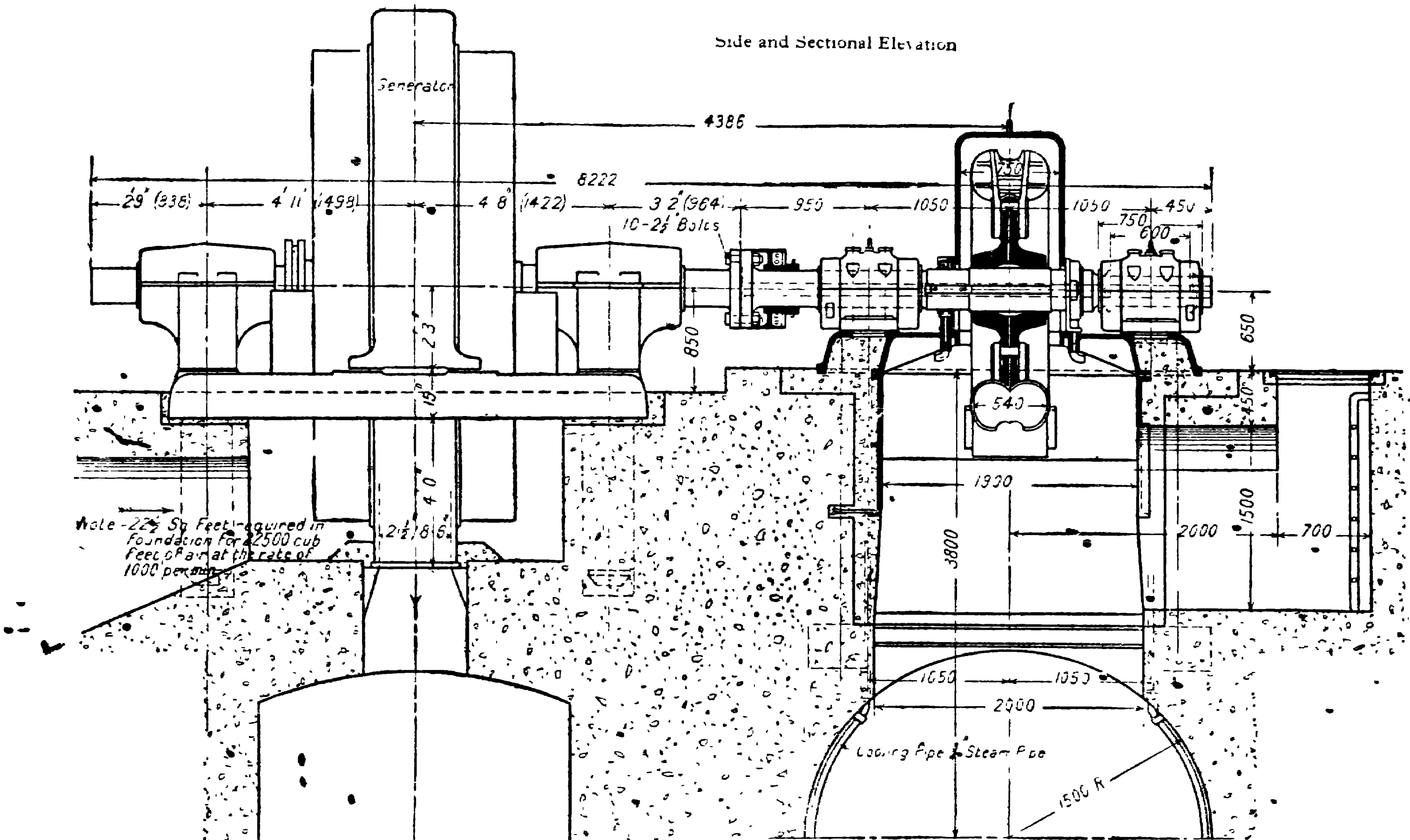
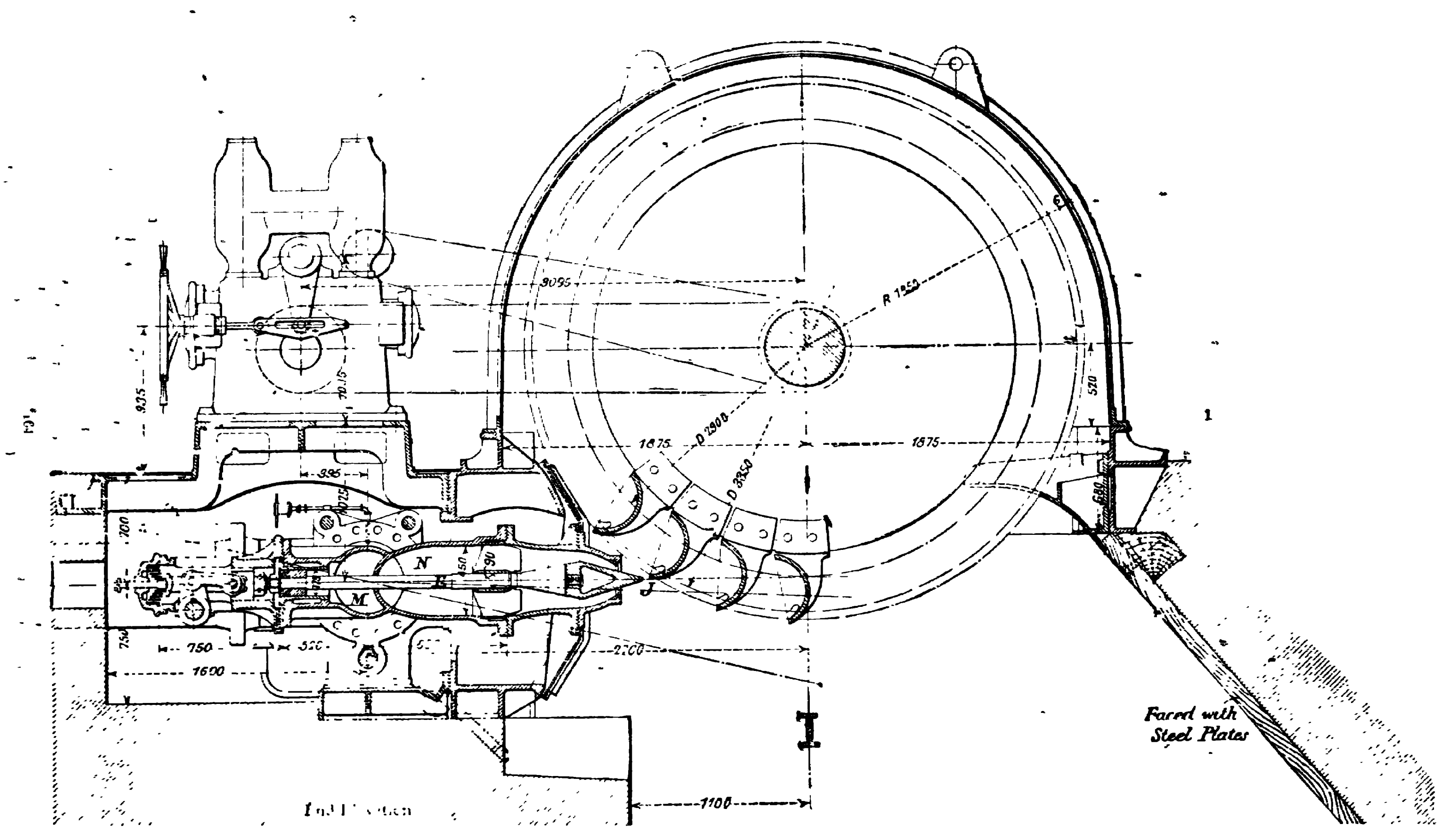


Fig. 119—8000-B H P. Pelton Wheel Head = 1015 ft.; Speed 375 r p m. Tasmanian Government Power Station Extension


Faced with Steel Plates
R 1950
D 2900
D 3350
3095
1875
1875
520
680
2200
1100
935
1015
1025
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750
1600
N
M
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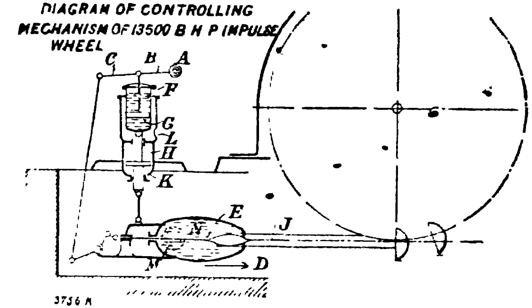
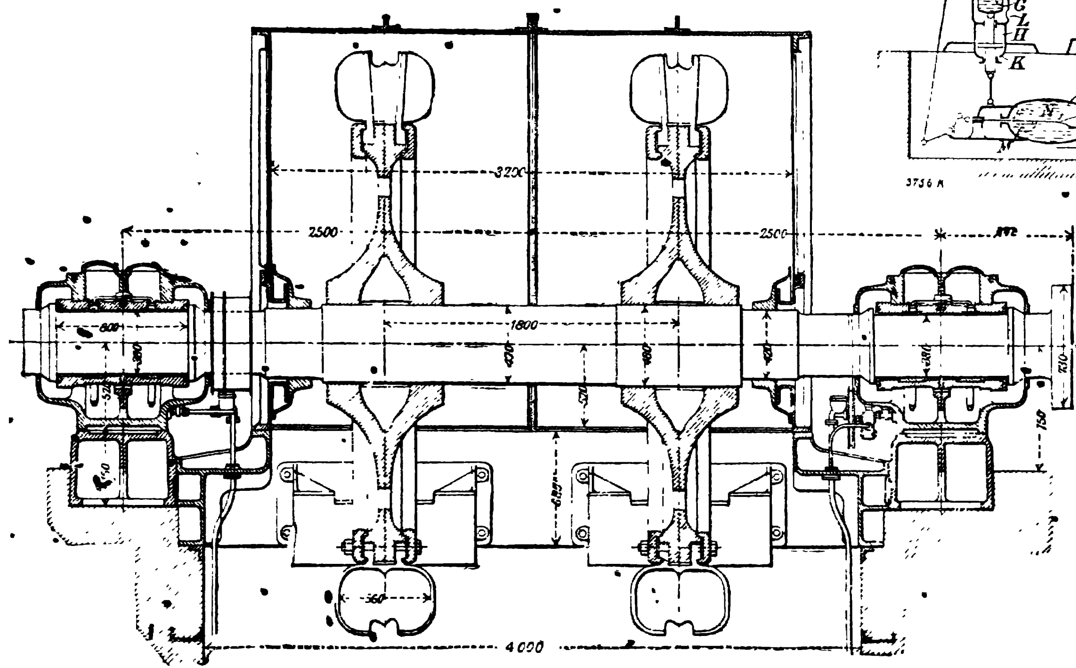




Sectional Elevation

Fig 120 —13 500-h p Pelton Wheel at Tyssedal Power Station. Head = 1330 ft, 250 r p m Pitch circle diam = 9 52 feet

2. Torsional deflection in the operating shafts renders it difficult to ensure equal gate openings on all runners.

3. The cost of the substructure is usually greater than in the case of a vertical single-runner turbine.

4. Owing to shock and interference between the discharge streams from the runners of a double-runner turbine, a greater proportion of the kinetic energy of discharge is lost than in the case of a single-runner turbine. This effect is increased by the impossibility of avoiding sudden changes of direction of flow in the water leaving the casing of the double turbine. Since a high velocity of discharge is a feature of a high specific speed turbine, this factor then becomes of special importance.

With a single-runner unit, on the other hand, only one gate mechanism is necessary, and this is outside the turbine casing and accessible for inspection at all times. Repairs to this can be carried out without dismantling the turbine. A long tapering draft tube, without any sudden changes of direction, can be used, and it becomes possible to adopt the type of moulded volute construction shown in fig. 146. Development, especially in the United States, has of recent years been tending to a more general use of this vertical single-runner type of unit for medium and even high heads, as well as for low-head plants, except where the general arrangement calls for an overhead intake to the turbines, or where direct-current generators are to be installed.

91a. Nominal Diameter of a Turbine Runner.—In a low-speed reaction turbine of the types shown in fig. 97*a* and *b*, the vane edges at the point of entry are parallel to the axis of the shaft. The diameter is measured at this point, and has a perfectly definite value. In a high- or medium-speed wheel the inlet edges of the vanes are usually inclined, while the discharge edge of the bucket has a maximum diameter which is greater than any point on the inlet side. In this case there is no definite convention as to what point shall be taken as giving the diameter of the wheel, and this discrepancy accounts to some extent for the fact that a turbine given as a certain diameter by one maker will have a greater capacity, speed, or output than one nominally of the same diameter quoted by another maker.

The diameter at the points 1, 2, 3, and 4 of fig. 118 is given as the nominal diameter by different makers. Here (2) is the mean diameter at inlet. The diameter shown at (4) is most commonly given. In any estimate involving the diameter of the runner, the prospective buyer should satisfy himself at which point this is measured.

Limiting Sizes of Turbine Runners.—At the present time the maximum diameter of runner for which any turbine maker will quote is about 17 ft. 6 in. A runner of this size and of normal design will give about 5900 b.h.p. under 20-ft. head, or about 16,500 b.h.p. under 40-ft. head, the variation in output with head being sensibly proportional to $H^{\frac{3}{2}}$.

92. The Pelton Wheel.—Except for very large units the Pelton wheel

is seldom constructed with a vertical shaft, owing to the advantages of simplicity of construction and facility for inspection afforded by the horizontal shaft type. In general only one nozzle is used on a single wheel. Two jets have been used in a number of cases, and in this way the power can be practically doubled. Such jets are usually placed approximately at right angles, as shown in fig. 119.* It is found, however, that in a

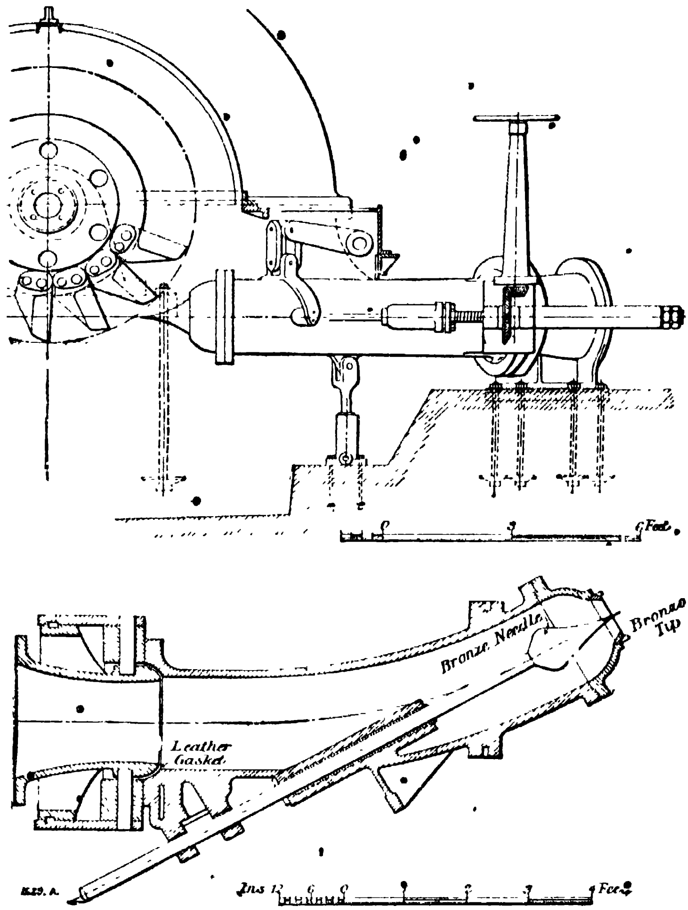


Fig. 121.—Deflecting Nozzle with Hand Adjustment to Needle

horizontal shaft unit the splash from one jet affects the other, and reduces the efficiency somewhat. Where the output required from a unit is greater than can be obtained from a single jet, it is usually preferable to mount two single-jet wheels side by side on the same shaft (fig. 120).†

Nozzles.—The modern Pelton wheel is always fitted with a circular

* *Engineering*, 2nd July, 1920, Messrs. Boving & Co., Ltd., London.

† *Engineering*, 28th August, 1914, Messrs. Escher Wyss & Co., Zurich.

nozzle, with an axial needle or spear for regulating the size of the jet. Other shapes of jet have been used, but all such forms suffer a greater windage loss than the circular. Also all other forms tend to become circular, and in the process tend to become unsteady. The maximum practicable diameter of jet appears to be about 12 in. The axial position of the needle in the nozzle is regulated either by hand (fig. 121), or, in all important installations, by the governing mechanism, as shown in figs. 119 and 120. For high efficiencies the diameter of the pitch circle of the buckets should not be less than about twelve times the diameter of the jet.

Buckets.—The original Pelton buckets were of rectangular section (fig. 122*a*). These have been superseded by the elliptical bucket (fig. 122*b*), in which that part of the lip in the line of the jet is omitted. The lip and ridge of the original bucket deflect the jet in two planes at right angles, and as the paths of the streams thus formed cross, a certain amount of energy is dissipated by their impact. Also the lip tends to deflect the jet radially inwards towards the rim of the wheel, in which case some fouling of the succeeding bucket is inevitable. The sharp curves and corners of this type of bucket cause an appreciable loss in eddy formation, and tests show that the efficiency obtained with the modern form of bucket is from 6 to 10 per cent greater than with the older form.

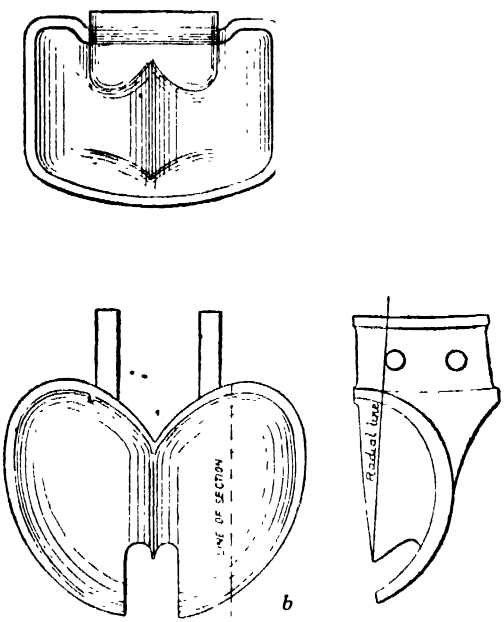


Fig. 122

The angle through which the jet is deflected by the bucket should be as nearly 180° as possible. In order that the discharge from one shall clear the back of the following bucket, in practice this angle is limited to a maximum of about 165°.

The friction loss in the buckets increases with the wetted area, and to reduce this the number of buckets should be as small as is consistent with continuous impact, while they should be made no larger than is necessary to give the required change of direction with easy curves and without shock. The surface should be as smooth and well finished as possible. In modern practice the width of the buckets is between three and four times the jet diameter, the ratio diminishing as the size of jet is increased.

In a high-speed wheel the runner consists of a steel disk, to which the buckets are bolted. The latter usually carry two lugs which straddle the wheel rim, and are attached to it by two bolts. In some cases a double-wheel disk has been used with the buckets carrying three lugs. Two of these straddle the disks, and the third fits between the disks. The lugs are so designed that a single bolt passes through the rear lugs of one bucket

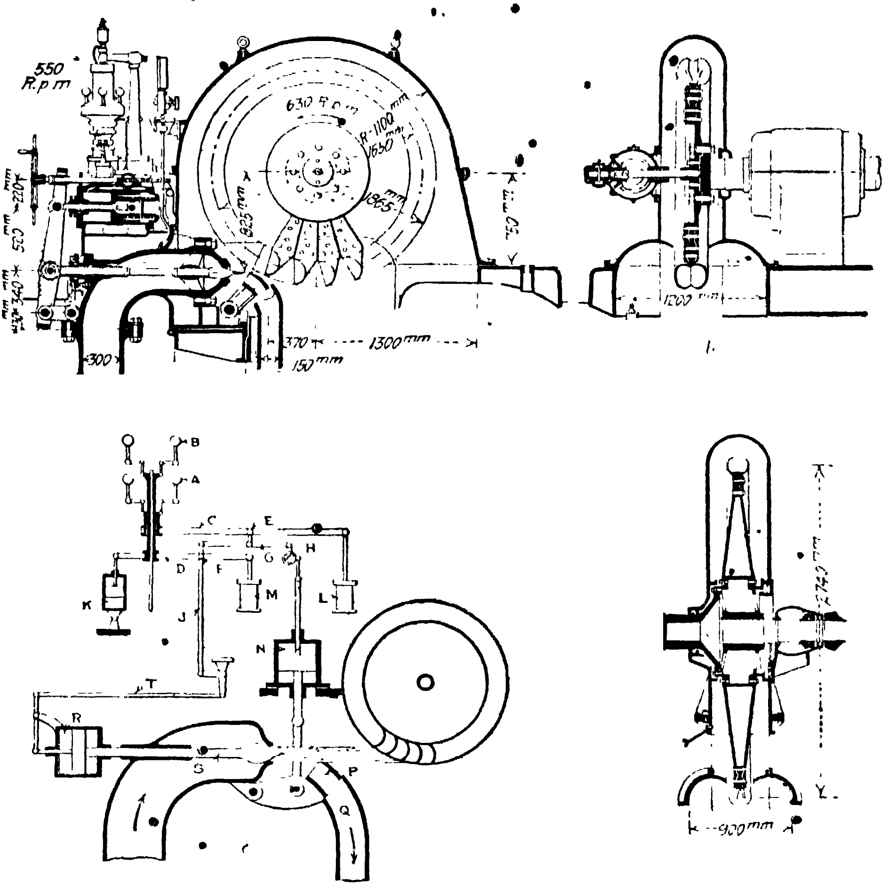


Fig 123 --Arniberg Power Station 3000 h p , 2800 ft head, 630 r p.m Pitch circle diam 5 41 ft

and the forward lugs of the next. This type of construction is advantageous where large buckets are to be fitted to a wheel of comparatively small diameter. For exceptionally high peripheral speeds it is inadvisable to rely on bolts to resist the large centrifugal forces, and one type of construction which has been successfully used under such conditions is shown in fig. 120. Here each bucket carries two lugs which closely straddle the disk, and are kept in position by two fitting bolts and by two steel rings, one on each side of the wheel.

Fig. 123*b* and *d* shows a type of construction adopted at the Arniberg

Power Station. Here units developing respectively 3000 h.p. and 1300 h.p. under 2800 ft. head have wheels of about 5 ft. 5 in., and 9 ft. 6 in. diameter, and make respectively 630 r.p.m. and 360 r.p.m. In both these wheels rivets and not bolts are used, so that individual buckets cannot be replaced.

In an installation at the Fully Hydro-electric Station in Switzerland, operating under the extremely high head of 5412 ft., the velocity of the buckets is 303 ft. per second. In this case the buckets are of forged steel. Each is provided with two lugs which fit into a wedge-shaped slot in the forged-steel disk (fig. 124).* A series of openings are made in the periphery

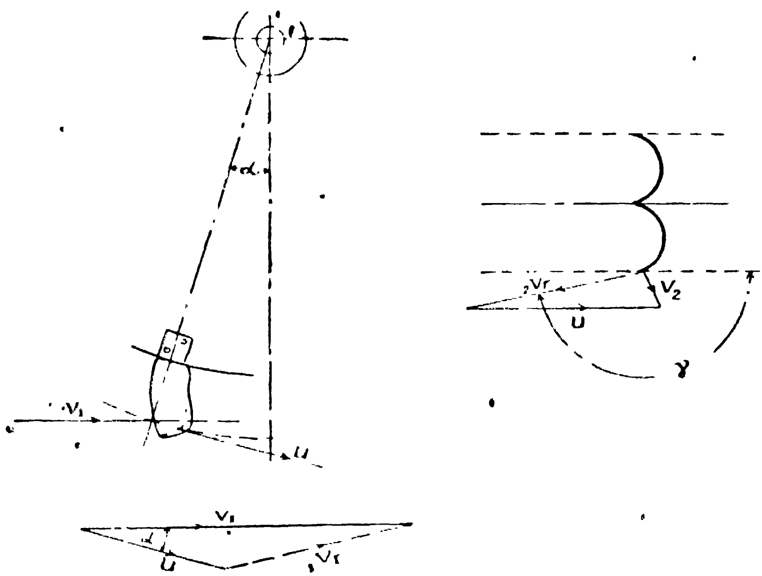


Fig 125

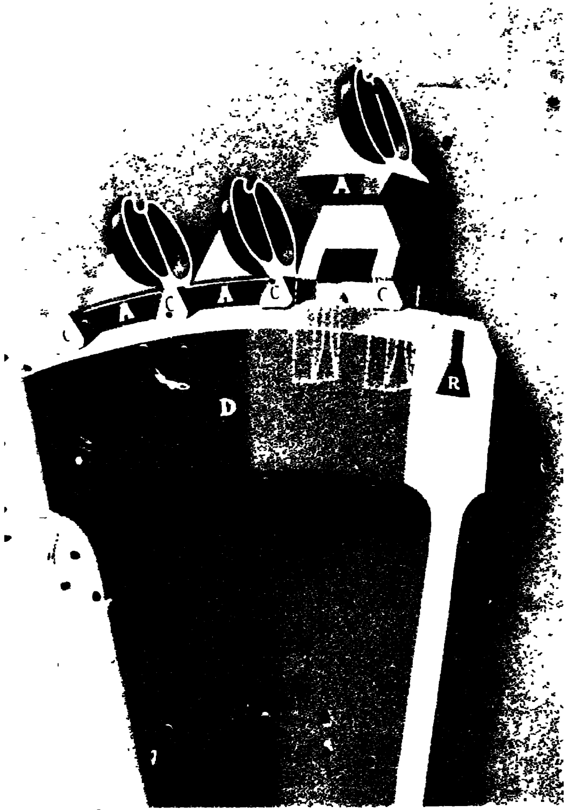
of the wheel through which the lugs are inserted, and the required spacing is obtained by distance pieces of triangular section inserted between the buckets. The last distance piece is inserted while the rim is heated, and the contraction on cooling binds the whole firmly together.

93. Hydraulics of the Pelton Wheel.—If H be the pressure head behind the nozzle of a Pelton wheel, the velocity of efflux is equal to $C_v\sqrt{2gH}$ ft. per second, where C_v, the coefficient of velocity, in a well-formed needle nozzle is approximately ·99. Calling V_1 this velocity, the horse-power of the jet is equal to

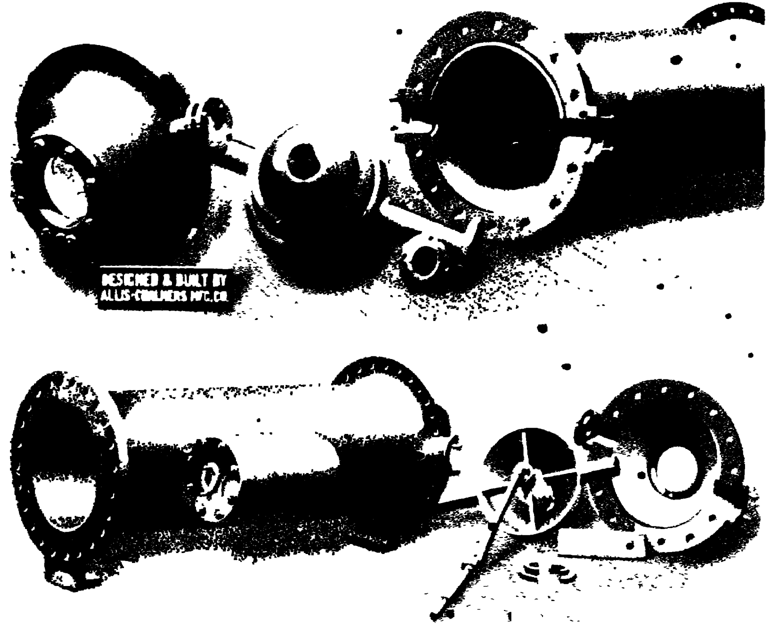
$$\frac{62{\cdot}4aV_1^3}{550 \times 2g} = {\cdot}00176aV_1^3 \text{ h.p.}$$

where a is the area of the jet in square feet.

* By courtesy of Messrs. Piccard, Pictet, et Cie, Geneva, and of Messrs. Vickers, Ltd.



124.—Method [illegible] Attaching Buckets in the
Fully Pelton Wheel Installation



F[illegible] 125.—D[illegible] N[illegible] Pelton

Let u = peripheral speed of buckets at pitch circle.
,, V_2 = final absolute velocity of water leaving the buckets.
,, ${}_1v_r$ = relative velocity of jet and bucket at entrance.
,, ${}_2v_r$ = relative velocity of jet and bucket at discharge.
,, α = mean angle between jet and tangent at point of contact.
,, γ = total angle of deflection of jet. (Fig. 125).

Then the initial velocity of jet in direction of tangent at point of impact $= V_1 \cos \alpha$.

The component, parallel to the tangent at discharge, of final velocity relative to bucket $= {}_2v_r \cos \gamma$.

$\therefore$ Absolute velocity in this direction at discharge $= u + {}_2v_r \cos \gamma$.

$\therefore$ Change of tangential momentum per second, per pound

$$= \frac{1}{g}\{V_1 \cos \alpha - u - {}_2v_r \cos \gamma\}.$$

$\therefore$ Work done per pound of water per second

$$= \frac{u}{g}\{V_1 \cos \alpha - u - {}_2v_r \cos \gamma\} \text{ ft. lb.}$$

$$\therefore \text{Efficiency} = \frac{u}{gH}\{V_1 \cos \alpha - u - {}_2v_r \cos \gamma\}.$$

The loss due to friction and eddies in the buckets $= \dfrac{{}_1v_r^2 - {}_2v_r^2}{2g}$ ft. lb. per pound, where ${}_1v_r = \sqrt{V_1^2 + u^2 - 2V_1 u \cos \alpha}$.

The loss due to rejection of kinetic energy in the discharge $= \dfrac{V_2^2}{2g}$ ft. lb. per pound, where $V_2^2 = u^2 + {}_2v_r^2 + 2u_2v_r \cos \gamma$.

Tests show that in an average wheel ${}_2v_r$ may be as low as from ·6 to ·7 ${}_1v_r$. In a well-designed bucket, however, having a ratio of bucket width to jet diameter not less than about 3·3, this ratio approximates to ·80 or even ·85. If the angle of deflection were 180°, and if the buckets were frictionless, the value of the peripheral speed of the wheel for maximum efficiency would be $V_1 \cos \alpha \div 2$, or approximately $V_1 \div 2$, since α is small. When account is taken of the loss in the buckets and of the fact that γ is less than 180°, the best peripheral speed lies between ·44 and ·48 V_1, the higher ratio being possible with the most efficient buckets.

Taking $\alpha = 10°$, $\gamma = 165°$, $u = ·46V_1$, ${}_2v_r = ·82 {}_1v_r$, this makes ${}_1v_r = ·553V_1$, ${}_2v_r = ·453V_1$, and the hydraulic efficiency becomes 88·6 per cent, a value agreeing closely with the results of good modern practice. In this case the loss in the buckets is 10·1 per cent, and at discharge 1·3 per cent.

Specific Speed.—As for a reaction turbine, the specific speed of a Pelton wheel is given by

$$N_s = \frac{N\sqrt{P}}{H^{\frac{5}{4}}}.$$

But $N = \dfrac{60u}{\pi D}$, where D is the diameter of the pitch circle,

$$\begin{aligned}
\text{and } P &= \frac{\cdot 00176\pi d^2}{4} V_1^3\eta, \text{ where } d \text{ is the diameter of the jet} \\
&= \frac{\cdot 00176\pi d^2}{4}\eta(2gH)^{\frac{3}{2}} \\
&= \cdot 610 d^2 H^{\frac{3}{2}}, \text{ if the efficiency } \eta = \cdot 85. \\
\therefore \sqrt{P} &= \cdot 780 d H^{\frac{3}{4}}. \\
\text{Also } u &= \cdot 46\sqrt{2gH} \text{ (approx.)}. \\
\therefore N_s &= \frac{60 \times \cdot 46\sqrt{2gH} \times \cdot 780 d H^{\frac{3}{4}}}{\pi D H^{\frac{5}{4}}} \\
&= \frac{55d}{D}.
\end{aligned}$$

If D : d = 12, this makes the specific speed approximately 4·6. This is the highest value of the specific speed for a single-jet Pelton wheel for high efficiency. Thus the maximum speed obtainable with a single jet for an output of say 2500 b.h.p. under a head of 900 ft. would be given by

$$4\cdot 6 = \frac{N\sqrt{2500}}{(900)^{\frac{5}{4}}},$$

$$\text{from which } N = \frac{4\cdot 6 \times (900)^{\frac{5}{4}}}{\sqrt{2500}} = 455 \text{ r.p.m.}$$

With two jets, or two wheels each with a single jet, the maximum value with the same ratio of D : d becomes approximately 6·5. By reducing the ratio D : d to 9, the specific speed, with a single jet, becomes approximately 6·0. The full load efficiency of such a wheel is about 5 per cent less than that of a wheel with a specific speed of 5·0, but in order to obtain this efficiency the buckets require to be very carefully designed.

94. Speed Regulation of Pelton Wheels.—There are three general methods of regulating the speed of Pelton wheels. These are:

1. By deflecting the jet from the wheel by a deflecting nozzle or hood.
2. By a combined needle regulator and pressure regulator.
3. By a combined needle regulator and deflector, or deflecting nozzle.

In the first method the jet is deflected wholly or partly from the wheel by the action of a relay piston operated from the governor, or in small plants by the governor directly. The flow in the pipe line is independent of the load, no pressure surges are produced, and the method gives excellent governing. It is, however, very wasteful under a variable load,

and is only to be condoned where, as in some irrigation schemes, or in some installations on the higher reaches of a stream, the user is under an obligation to discharge water at a minimum rate for users lower down the stream. The loss may be reduced by using the deflector in conjunction with a hand-regulated needle. This is set by hand at intervals to give the maximum discharge likely to be required during the next period, and any fluctuation of load up to this maximum is handled by the deflector.

In the older forms of deflecting nozzle (fig. 121) * the nozzle is connected to the pipe line through a swivel joint packed by means of a leather ring, and is carried by the plunger rod of the relay cylinder. The pressure below this plunger and hence its position are regulated by a pilot valve operated by the governor. In one modified type (fig. 125*a*, in plate facing p. 170),† the nozzle casing is fixed. The needle, and a spherical dome in which is formed the jet orifice, are the only movable parts. The dome is carried on trunnions which are rotated by the governor mechanism, and fits the concave surface of the nozzle casing. The needle is provided with a universal joint inside the casing, and its stem projects through a stuffing box and carries a hand wheel for auxiliary regulation.

Fig. 120 shows a deflecting nozzle, in which the nozzle with the needle-regulating mechanism is mounted on a hollow trunnion through which the pressure water is conveyed. This is carried through a stuffing box which is easily kept tight. Instead of a deflecting nozzle, the jet may be deflected by means of a hood or deflector, fitted between the nozzle and the wheel, and whose position is regulated by the governor (figs. 123 and 126). This is lighter, cheaper, and gives less trouble in maintenance than the deflecting nozzle.

In the second method of regulation, the position of the needle is regulated by the governor, while to prevent pressure surges accompanying a sudden reduction of load a pressure regulator (p. 184) is fitted. While, owing to the smaller volumes of water to be handled, the pressure regulator is more suitable for Pelton wheels than for reaction turbines, it suffers from the liability of the valve to stick, and from the difficulty of ensuring its synchronous action, and in most recent plants the third method of regulation has been used.

Here both deflector and needle are regulated by the governor. The connection is so arranged as to allow of a rapid movement of the deflector on a reduction of load, followed by a slow movement of the needle closing the nozzle. When in its final position, the deflector is tangential to the reduced jet, ready for an immediate response to any fresh change of load. One arrangement of this kind is shown in fig. 126.‡ The needle carries a slotted extension G, and is forced to the right by a spring H, whose action is retarded by the action of the oil dashpot C. If the load is reduced the lever F is forced to the right by the servo-motor, deflecting the jet,

* *Proc. Inst Mech. Engineers*, January, 1910.

† By courtesy of Messrs. Allis Chalmers, Ltd.

‡ By courtesy of Messrs. The English Electric Co., Ltd , London.

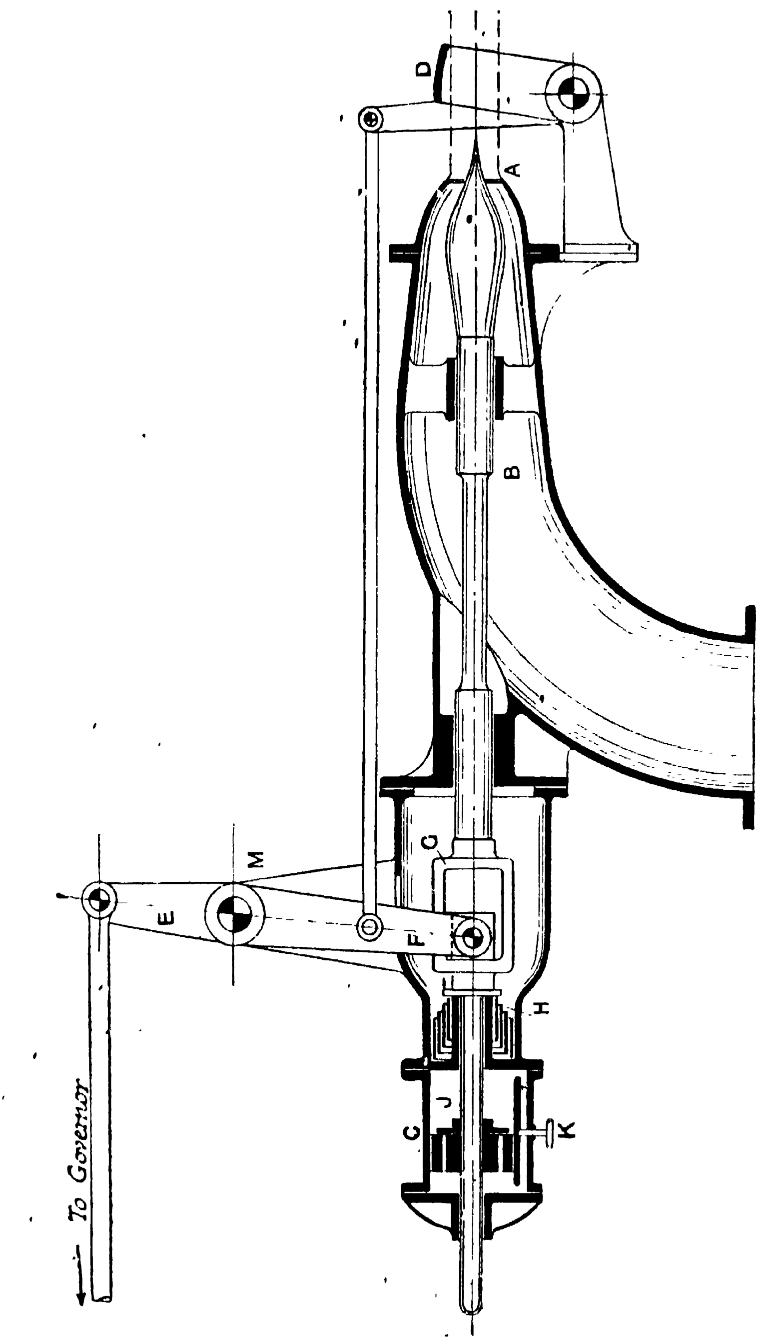


Fig 126

while the needle follows slowly until the slot is again in contact with the end of the lever F, with the needle tangential to the jet. On an increase in load the lever is forced to the left, and carries with it both the needle and deflector. The valves J in the dashpot plunger allow free motion in this direction.

Fig. 123*c* shows another method of obtaining the same result, as applied to the units of the Arniberg Power Station. Here the governor has two flyball systems acting on two levers A and B, with fulcrums at E and F. The fulcrum E is carried by the bar G, which has one end connected to the eccentric H, and the other to a rod J. The lever D is connected to a dashpot K, and the levers C and D control the regulating valves L and M respectively. The valve M controls the piston of the servo-motor R, which controls the needle S. The valve L controls the piston N and the deflector P. For a slow change of speed, levers C and D move in unison. The needle and the deflector are moved simultaneously, the proportions being so fixed that the deflector is brought finally into a position tangential to the jet. If the change of load is rapid, however, the dashpot prevents the free motion of the lever D. In consequence the deflector acts as before, but the action of the jet is much slower. When the size of the jet has been reduced to that corresponding to the reduced load, the deflector, through the agency of the rods T and J and the lever G with the eccentric H, is again brought tangential to the jet. The pipe line is 6850 ft. long, and varies in diameter from 18 to 24 in. Tests show that if 97 per cent of the full load is suddenly thrown on, the initial diminution of pressure is 11 per cent and the speed variation 17 per cent. When 97 per cent of the full load is suddenly thrown off, the initial increase of pressure is 5 per cent and the speed variation 5·5 per cent.

In the installation shown in fig. 120, a sudden reduction of load causes a rapid upward motion of the lever B, which raises the dashpot F and allows pressure water in the chamber H to escape through the opening at L. This reduction in pressure enables the pressure water in the chamber K to raise the piston in this chamber, and to deflect the jet from the wheel. The weighted dashpot then gradually falls, shutting off the discharge from H, equalizing the pressures in H and K, and bringing the nozzle back to its original position. Meanwhile the governor relay remains in the same position, and by the return movement of the nozzle the position of the needle relative to the nozzle is gradually altered, reducing the discharge. In this installation, when the full load is suddenly thrown off, the maximum variation in pressure is 12 per cent, and in speed 9 per cent. In the case of a gradual change of load, the dashpot and nozzle do not move, and regulation is performed by movement of the needle alone.

95. Runaway Speeds.—If, when load is thrown off the turbine, a failure of the governing mechanism or jamming of the gates causes the latter to remain open, the speed will increase considerably, and the rotors of the turbine and of the electric generators should be designed so as to be safe under the maximum runaway speeds which may be attained.

In a Pelton wheel installation, the maximum possible peripheral speed will be somewhat less than that of the jet, owing to mechanical friction and windage, and as the normal speed is slightly less than one-half that of the jet, the runaway speed should be taken as twice the normal speed.

With reaction wheels working under a constant head, and operating normally at the most efficient speed, the runaway speed is between 55 and 85 per cent above the normal speed. In a low-head turbine operating under a wide variation in the head, and designed for maximum efficiency under the average head, the runaway speed under the maximum head may be as much as three times the normal operating speed. This depends essentially on the range of head and on the design of the turbine, and in such an installation the runaway possibilities should be carefully investigated, with reference to actual tests on a wheel of similar design

Several types of over-speed regulators are available. These usually consist of a separate centrifugal governor operating an emergency control which, when coming into operation, shuts down the unit.

96. Selection of Turbine.—The choice of the most suitable type of turbine depends upon a number of factors, including the power, the head, the desired speed of rotation, and the special circumstances of the plant.

If the head is so large that its variations under varying conditions of seasonal flow can be neglected, the turbine can be considered from the view of constant head operation. If the head is low, the rise in level of the tail water in times of flood is usually much greater than that of the head water, and the percentage variation in head may become large. Under such conditions a turbine should be selected which will give satisfactory operation and reasonable efficiencies over the entire range of heads, and which will give high efficiencies under the highest heads, which occur when the minimum quantity is available. Some of the recent high-speed runners are extremely well adapted for such service.

Where the load variations are likely to be large, and where the quantity of water is limited, a turbine should be selected which will give its maximum efficiency under the average conditions of operation.

It is usually necessary to select a turbine having a given speed and output in order to operate a generator for which these characteristics are fixed. In such a case the turbine should be designed to give as nearly as possible the exact output required, so as to work at approximately full load, and hence under the most economical conditions. It is advisable to split up the installation into as few units as the operating conditions permit.

From a knowledge of the necessary output and speed, the characteristic speed is deduced from the expression,

$$N_s = \frac{N\sqrt{P}}{H^{\frac{5}{4}}}.$$

Having determined the characteristic speed, a type of turbine should be selected having a value of N_s, not less than the desired value, and, for high

efficiency, not greatly different from this value. If the calculated value for a single turbine is greater than is attainable with the type selected, the power must be divided between two or more units, until the required conditions are satisfied.

If, for example, an output of 20,000 h.p. be required at 150 r.p.m. under a head of 40 ft., the value of N_s for a single wheel would be 211. As this is higher than is attainable with any of the types in general use, it would be necessary to install more than one unit. The following table shows the number of turbines or of runners which would be necessary, with the corresponding specific speeds:

Number of runners ..	3	4	5	6	8	10
Specific speed ..	122	106	94	86	75	67

In making the final selection it is to be remembered that the efficiency, especially at part loads, of a turbine having a high specific speed is not so high as that of one having a lower specific speed. Also that a high-speed turbine requires a more carefully designed, and generally a more expensive, setting than does a low-speed turbine, so that it is necessary carefully to weigh the relative advantages of a reduction in the number and first cost of the units, against the possibilities of a higher efficiency.

Taking another case, in which 10,000 h.p. is required under a head of 600 ft. with a speed of 600 r.p.m., the value of N_s for a single wheel would be 20. In this case either a single reaction turbine, or two turbines each having a specific speed of $20 \div \sqrt{2} = 14{\cdot}1$ could be used, or a battery of Pelton wheels. Since the maximum specific speed of the Pelton wheel is about 5·0, the minimum number of wheels or of jets would be $\left(\frac{20}{5}\right)^2 = 16$, while if the speed of rotation could be reduced to 300 r.p.m., making the value of Ns for a single wheel equal to 10·0, only $\left(\frac{10}{5}\right)^2 = 4$ wheels would be necessary. In this case it would be a question of investigating the relative advantages and costs of an installation of four Pelton wheels, or of two double-jet Pelton wheels with generators designed for a speed of 300 r.p.m., and of two reaction turbines at 600 r.p.m. With water containing much grit in suspension the Pelton wheel installation, on account of the ease of renewal of the buckets and nozzles, would probably be found to have the balance of advantages.

These examples are probably sufficient to illustrate the influence of the factor of specific speed on the choice of the most suitable type of turbine.

The question as to whether a vertical- or horizontal-shaft machine, or one having a single runner or two or more runners, is most suitable depends largely on the special circumstances of the installation. These questions are discussed in some detail in Chapter X. The all-important factors

guiding the decision should be the reliability, the accessibility, and the efficiency of the unit. In the average hydro-electric installation the cost of the turbine itself and of its setting forms such a very small proportion of the total cost of the scheme, seldom exceeding 5 per cent of the total, that no question of its cost should be allowed to weigh against the advantages of freedom from breakdown, ease of repair, and efficiency of operation. It is to be remembered that any increase in efficiency enables the capacity of the head works, the head race, and the pipe line to be correspondingly reduced, and, with a limited quantity of water, enables just so much more electrical energy to be developed and sold. Generally speaking, the saving in these respects far outweighs any additional costs incurred by installing the most efficient turbine which can be obtained.

CHAPTER IX

Speed Regulation

Speed regulation; governors; pressure regulators; inertia of rotating parts; surge tanks; differential surge tanks; closed stand pipes; fluctuations of speed in practice.

97. Speed Regulation.—For a hydraulic turbine to be capable of close speed regulation under a varying load, it is necessary that it be provided with a reliable governor, that the inertia of the rotating parts be sufficiently great, and that the power conduit be so designed or so equipped that the rise or fall in pressure at the turbine, following any probable change of load, is only a small fraction of the working head.

As regards the latter factor, the easiest plant to control is one in which the turbine is installed in an open forebay fed directly from the supply canal. Here a demand for power is instantly met by an increased flow into the turbine with the velocity corresponding to the working head, and in this case, and also when the turbine gates are closed on a diminishing load, inertia effects are negligible. Both head and tail race should be of ample size, so that any fluctuation in flow may not cause an appreciable change in either level, while the approach channel should have easy curves and well-finished surfaces in order to reduce the possibility of any periodic wave formation.

Wherever possible the use of a long pipe line should be avoided, for it may be taken as generally true that the difficulties of governing increase as the slope from open head water to tail water is diminished.

Where it is necessary to supply the turbine through a pipe line, the force necessary to produce the required acceleration of the supply column on an increasing load, and its retardation on a diminishing load, can only be obtained by a fall or rise in pressure at the turbine. If *l* ft. be the length

of the conduit, if v_1 and v_2 ft. per second are the velocities of flow required to give the initial and final steady loads, and if t sec. be the interval of time over which acceleration takes place, then, if the acceleration were uniform, the drop in pressure at the turbines below that obtaining with steady flow would be given by

$$p = \frac{v_2 - v_1}{t} \times \frac{l}{g} \text{ ft. of water.}$$

More generally, if δv be the change of velocity in time δt, we have

$$p = \frac{l}{g}\frac{\delta v}{\delta t} \text{ ft.}$$

Since, for a given percentage increase of load, δv is proportional to the mean velocity of flow, the corresponding drop in pressure will be proportional to this mean velocity, so that, other things being equal, the difficulty of governing increases with the working velocity adopted for the pipe line.

With a long pipe line the magnitude of these inertia effects would be too great to permit of successful governing, and to reduce them some form of pressure-regulating device is necessary. The form which this should take depends to some extent on the type of installation.

If the pipe line is steep, with a gradient exceeding about 45°, a demand for energy on an increasing load receives a sufficiently rapid response, and speed regulation in this direction, with moderate velocities of flow, is comparatively easy. The increase of pressure on a diminishing load must, however, be guarded against by the provision of a surge tank, relief valve, or automatic pressure regulator. The relief valve (Art. 79) consists of an automatic valve which is adjusted so as to open outwards when the pressure exceeds the normal by a few pounds per square inch The pressure regulator consists of a valve usually operated through the governing mechanism, and so arranged as to open immediately the speed of the turbine increases, and to close very gradually (Art. 99).

If the head is not too great to allow it to be constructed at a reasonable cost, a surge tank affords the most satisfactory solution, since it also reduces the inertia effects on an increasing load. Where the pipe line is long and the gradient comparatively small, a surge tank is essential for governing the turbine on an increasing load. If the head is too great to permit of an open surge tank, a closed tank may be used. Without such a device it becomes impossible to obtain close speed regulation of a reaction turbine, except by the provision of an abnormally heavy fly-wheel, and in an extreme case the only installation capable of giving satisfactory operation consists of a Pelton wheel fitted with a deflector hood, or deflecting nozzle, discharging full bore at all loads except as regulated from time to time by hand.

In the case of a low-head plant fed through a long closed pipe line, care should always be taken to ensure that the drop in pressure accom-

panying the maximum acceleration possible in practice is not so great as to reduce the pressure appreciably below that of the atmosphere. This condition often renders the use of a surge tank at the power house essential.

From what has been said, it will appear that the surge tank forms a most important feature of many hydraulic installations, and that by its use many schemes of development become possible which would otherwise be impracticable. Its possibilities are discussed in greater detail in Arts. 102–6.

The characteristics of the turbine also have an important bearing on its speed regulation. Where the turbine efficiency falls off appreciably between three-quarters and full load, as may be the case, the difficulty of close regulation over this range of loads is greatly increased, since an increase of load requires a greater additional supply of water than would be the case with a constant or an increasing efficiency.

98. Governors.—The requirements of a successful governor are:

1. Sensitiveness; i.e. it should alter the position of the turbine gates with as small an increase of speed as possible. With a good hydraulic governor, a fluctuation of speed not exceeding 0·5 per cent is usually adequate to produce motion of the gates.

2. Rapidity of action; i.e. it should be sufficiently powerful to move the gates rapidly. In general the quicker the response on an increasing load the better. On a falling load, however, a too rapid response tends to set up water-hammer, except in the case of a Pelton wheel with jet deflector or deflecting nozzle. No definite rule can be laid down for the best speed of operation. A greater speed is possible with a short penstock and low velocities of flow than with a longer penstock or higher velocities. In general a governor which will give the full range of gate opening within two or three seconds will satisfy all requirements; but, if necessary, the modern governor is capable of reducing this time to one second or even less.

Owing to the inertia and considerable frictional resistance of the turbine gates, the centrifugal governor is not sufficiently powerful in itself to give the required motion, and some form of relay becomes necessary. In the earlier turbines various types of mechanical relay were used, but while for such purposes as driving textile machinery, where the changes of load are relatively small, this type fulfils the requirements, it has proved inadequate to satisfy the more exacting requirements of electric generation. For such a purpose the pressure-operated relay provides the only satisfactory solution. Here the centrifugal governor operates a regulating valve which admits either oil or water under pressure to one side or other of a piston in a relay cylinder, this piston being connected to and operating the gate mechanism. The governor is usually rated in foot pounds, according to the work done per stroke of the servo-motor or relay cylinder, at a given supply pressure. The operating fluid may consist either of water or of oil. If water, multi-stage centrifugal pumps may be used for producing the pressure; if oil, plunger pumps or gear-wheel pumps are used. The pumps deliver into a pressure tank provided with a large air chamber,

which enables the draft on the pressure supply during the short periods of gate motion to be made at a greater rate than that of the pump supply. Owing to the possibility of sticking due to grit in the water and to corrosion, the water-pressure governor has been generally displaced, except in the largest systems, by the oil-pressure governor. In the case of a large system the use of water, with the addition of soluble oil or glycerine, has some advantages in cheapness and because it enables the more convenient form of high-speed centrifugal pump to be used. In a large station, a central supply system with motor-driven pumps may be used, each turbine having its own pressure tank close to its relay cylinder or servo-motor, in order to reduce inertia effects. While this is less costly than an independent direct-driven pump for each turbine, it has the disadvantage that any failure of the supply may involve shutting down the whole of the units involved.

In units of small and medium size, the governor control mechanism, the servo-motor, and the oil pump are usually incorporated. In modern large units the servo-motor is often incorporated in the design of the turbine. The governor and its control mechanism form a separate unit, and are located in the most convenient situation, which may be some distance from the turbine. By having an emergency pipe connection between adjacent sets, and designing each main oil-pressure set of sufficient capacity to operate two main units, the danger of failure is greatly reduced, and this system appears to afford a satisfactory solution of the problem.

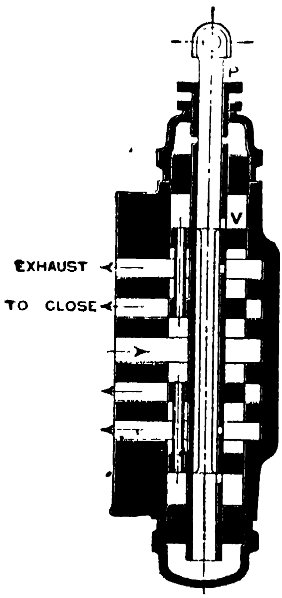


Fig 127 —Balanced Regulating Valve for Servo-motor Cylinder

Fig. 127 * shows a typical regulating valve for the oil-pressure system. Here the governor operates the small pilot valve P, which, when raised or lowered from its central position, relieves the oil pressure at the top or bottom of the balanced relay valve V, and by raising or lowering the latter admits pressure oil to one side or other of the piston of the servo-motor. For smaller units the relay valve may be operated directly from the governor.

3. Stability; i.e. the governor should bring the gates to their proper position with as little hunting or as few oscillations as possible about this position. To ensure this, some form of relay return or compensating device is usually necessary. The reason for this will be evident if it be considered that, as the speed falls, the gates are opened and the supply column is accelerated, this action persisting until the supply of energy per unit time is equal to the demand. But the acceleration of the water column goes on for an appreciable time after the gate opening has ceased, and in

* Bergstrom, *Proc. Inst. Mech. E.*, 1920.

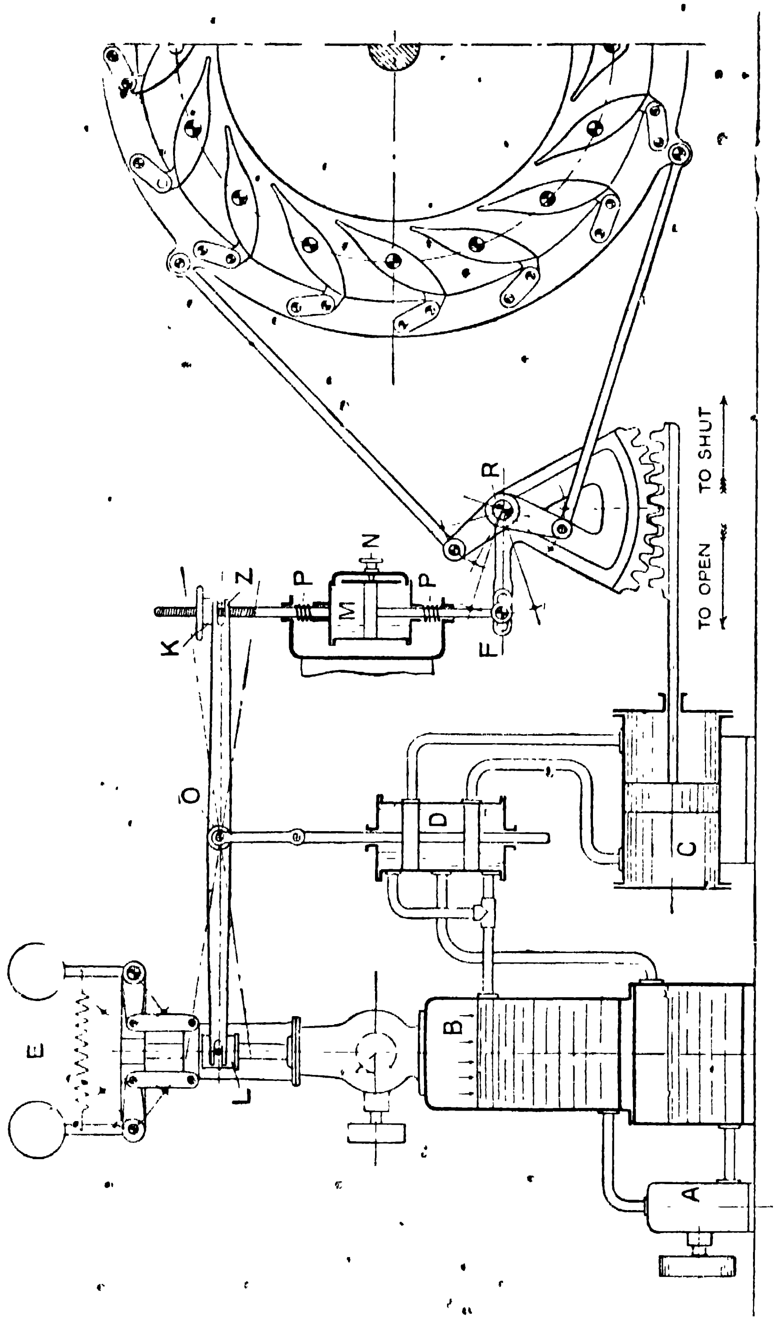


Fig 128 —Governing Arrangement for Francis Turbine

consequence the supply becomes too great for the requirements of the wheel, the speed increases, and the governor commences to close the gates. This checks the motion of the supply column, and in virtue of its inertia produces an increased pressure at the valve, and a temporarily increased

velocity of flow through the gates. The speed of the wheel thus increases still further, and the gates are closed until an instantaneous balance is again set up between supply and demand. As the inertia pressure falls, the supply now becomes less than the demand, the speed falls, the gates commence to reopen, and the state of hunting which is here outlined may not die out for some considerable time. To prevent hunting, it is necessary that the governor should cause the gates to overrun their final position slightly, and to bring them back slowly to their final position, corresponding to the altered load. This latter operation is performed by means of some form of compensating device.

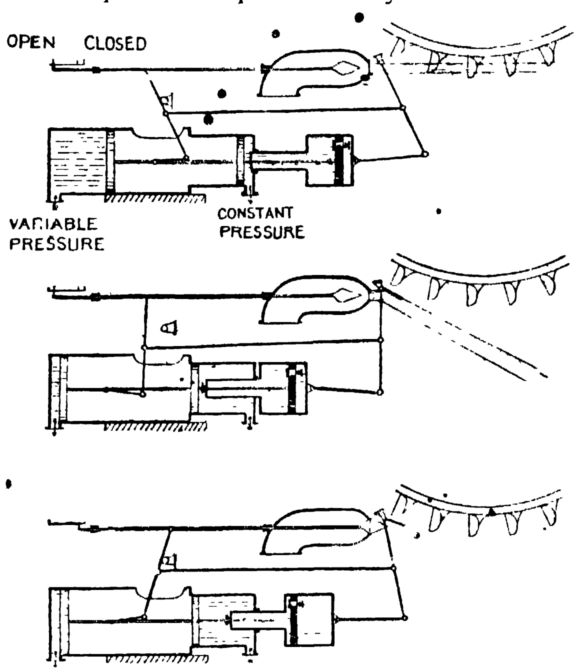


Fig. 129.—Governing Arrangement for Lac Fully Installation

Fig. 128 shows diagrammatically one such typical arrangement,* where A represents the oil pump, B the pressure-oil receiver, C the servo-motor, D the distributing valve, and F the compensating device. The governor is so adjusted that at normal speed the sleeve L and the distributing valve are in their central positions as shown. An increase in speed lifts the valve about Z as a fulcrum, forcing the servo-motor piston to the right and closing the gates. This motion lowers the points F and Z, bringing the distributing valve back to its central position and preventing any further motion of the piston of the servo-motor. The distance between F and Z can be altered by hand so as to enable the speed to be varied slightly while running if required. In this arrangement each position of the guide vanes corresponds to a definite speed of rotation. In order to obtain a sensibly constant speed at all loads, a spring-loaded oil dashpot M is introduced between the points F and Z. On any rapid motion of the gates following a sudden change of load, this dashpot acts as a rigid connection and the controlling springs PP are extended or compressed. Under the action of these springs, the dashpot cylinder is gradually brought back to its central position in its guides, while its piston remains stationary. This brings the point Z back to its central position. The accompanying motion of the distributing valve

* Messrs. The English Electric Co., Ltd.

causes the speed either to increase or diminish slightly until L attains its central position corresponding to constant turbine speed.

Similar devices, as applied to Pelton wheel installations, have already been illustrated in figs. 120 and 123, and fig. 129 shows diagrammatically the mechanism adopted for the governing of the Lac Fully installation.

Fig. 130 * shows an arrangement of oil-pressure governor in connection with a Francis turbine, while fig. 131 † shows the governor used in connection with, and shown diagrammatically in the sketch relating to the installation of fig. 128. In all modern governors hand-wheel operation is installed for starting up the turbine, and for emergency use in case of a breakdown to the governor.

99. Pressure Regulators.—In high-head plants where the pipe line is long and circumstances do not permit of the provision of a surge tank, the governor may be operated in conjunction with a pressure regulator or relief valve arranged so as to open and discharge water into the tail race as the turbine gates close. One such regulator as applied to a Francis turbine is shown diagrammatically in fig. 132. A rapid closure of the gates lifts the weighted dashpot A, and along with it the relief valve. The valve then gradually closes under the action of this weight, the time of closure being regulated by the amount of opening of the needle valve, which controls the rate of flow of the oil from one side to the other of the piston B.

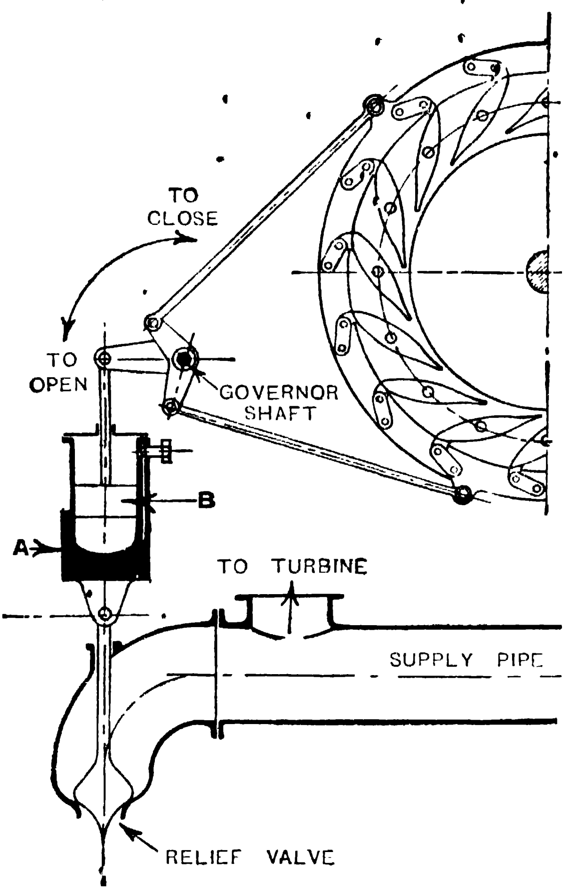


Fig. 132

Fig. 133 ‡ shows the relief valve used on the 20,000 h.p. units of the Shawenegan Water and Power Co. Here penstocks 600 ft. long and

* By courtesy of Messrs. Gilbert Gilkes & Co., Kendal.
† By courtesy of Messrs. The English Electric Co., Ltd.
‡ Department of the Interior, Ottawa: *Water Resources*, Paper No. 17, 1916.

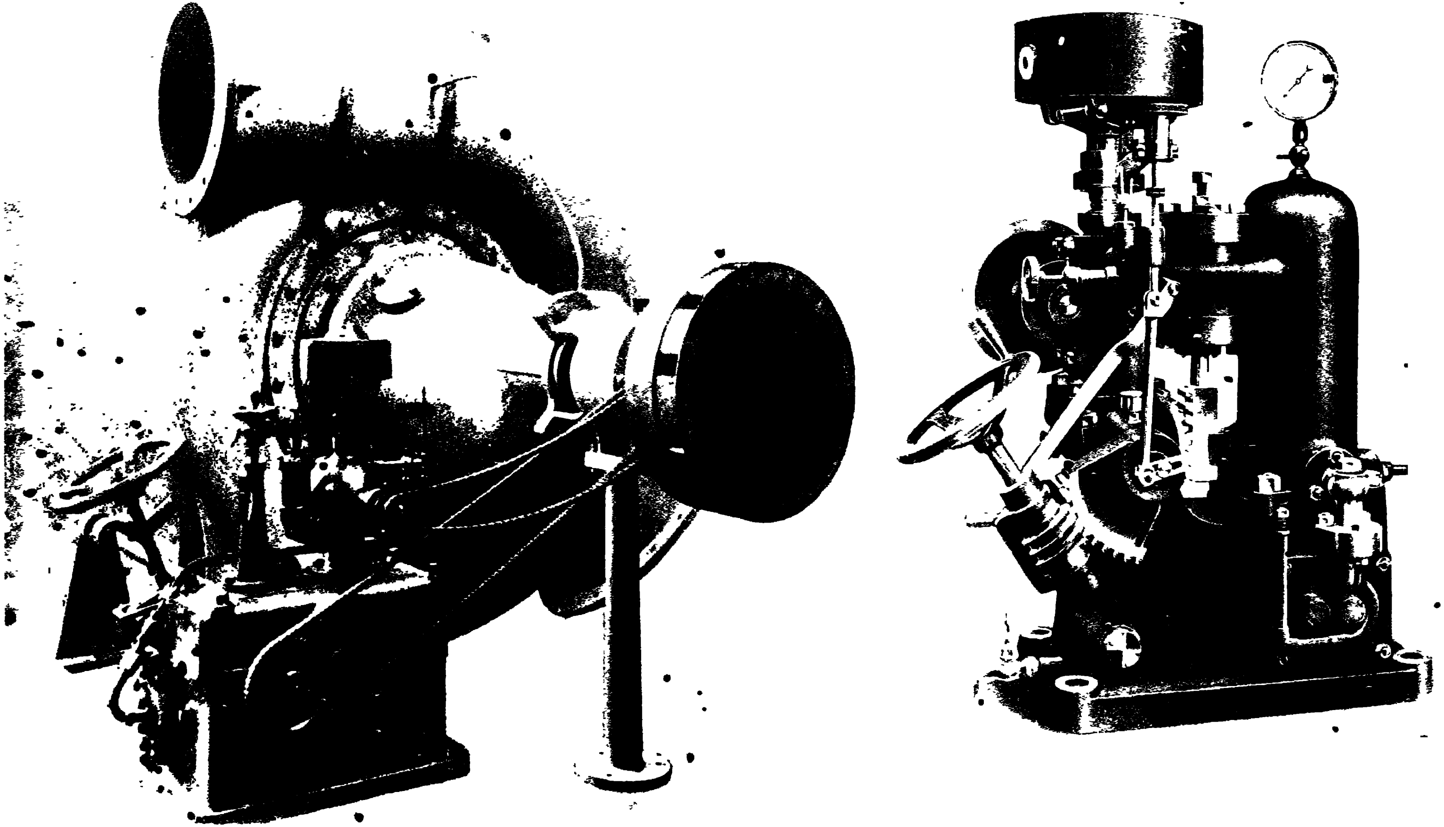


FIG. 130.—ARRANGEMENT [illegible]
SPIRAL-CASED FRANCIS TURBINE

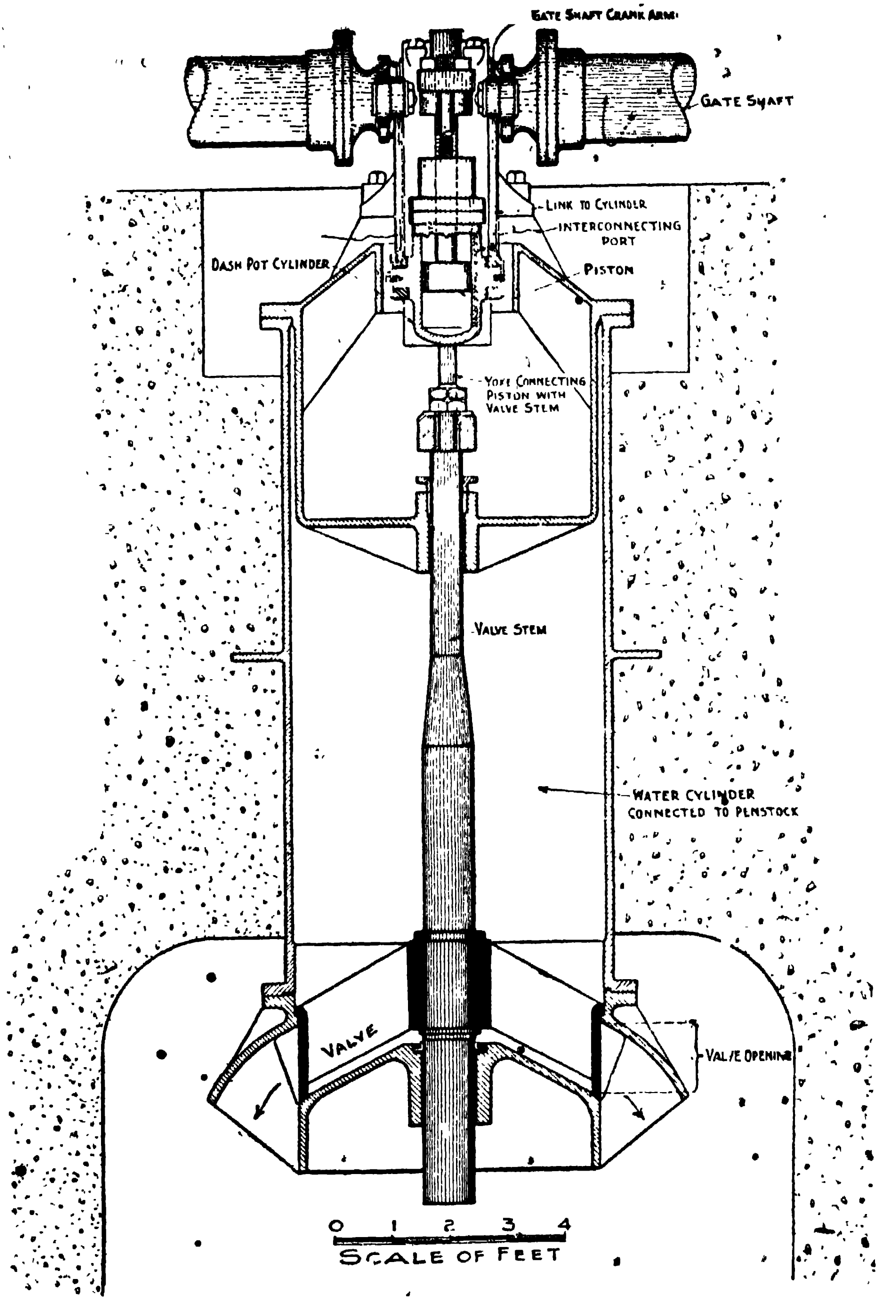


Fig 133.—Relief Valve for 20,000 h p units of Shawenegan Water and Power Co

14 ft. in diameter, with a normal velocity of 8·5 ft. per second under 145 ft. head, supply each turbine. Each penstock divides into two branches, one for each of the wheel cases of the turbines. The relief valve is set in

the notch between the branches, and discharges into the draft tube. The operating gate shaft is connected to the relief-valve dashpot, and the dashpot piston to the relief-valve spindle by yokes and trunnions. The dashpot is oil-filled and its ends are interconnected by two bypasses cored in the casting, one bypass containing a needle valve and the other a spring check valve operating in only one direction. When the turbine gates are closing, the dashpot is raised by the gate shaft lever. Should the speed of this movement be such that the oil underneath the dashpot piston will bypass through the needle valve to the other side of the piston without building up sufficient pressure to overcome the weight of the relief valve, this valve remains closed, while if the movement occurs at such a rate that the pressure is sufficient to overcome the weight of the relief valve, the valve is opened, and closes again gradually by return flow of the oil through the check valve.

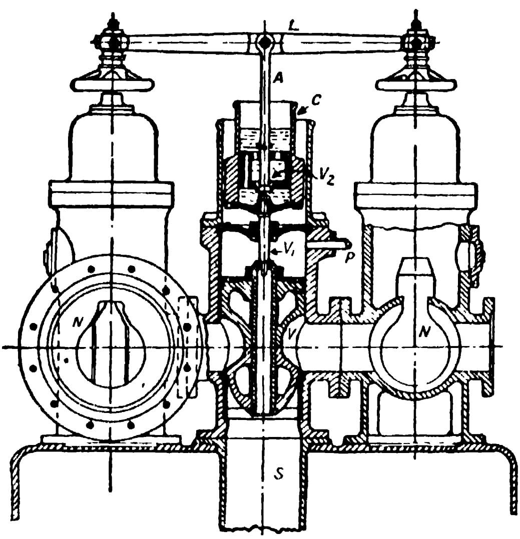


Fig. 134

In a large unit the mechanical operation of a large relief valve necessitates a very powerful servo-motor. To obviate this the type of regulator shown in fig. 134 as applied to a twin Pelton wheel is used. Here the lever L is connected to the piston rods of the two relay cylinders, and carries the dashpot rod A. The dashpot C is weighted and carries the needle valve V_1. The relief valve V is slightly overbalanced hydraulically so as to remain closed under normal conditions, and pressure water supplied through the small pipe P keeps it closed as long as the valve V_1 is closed. If this valve is opened, however, by a sudden upward motion of L, the pressure above the main valve is reduced, and the valve opens, afterwards closing gradually as the dashpot falls by its own weight. The valve V_2 permits of a sudden depression of L without unduly straining the dashpot rod A.

Fig. 135 * shows another type of pressure regulator in which the operating force is provided by the pressure of the supply water. The lever A is coupled to the gate mechanism, and a closure of the gates moves this lever from right to left. This lowers the dashpot B, and, if the motion is sufficiently rapid, carries down the dashpot piston, lowering the end C of the lever L, and opening the pilot valve D. This allows pressure water to escape from the chamber E, the pressure on the relief valve F overcomes the upward pressure on the balance piston G, and the

* The Wellman Seaver Morgan Co.

valve opens. The pilot valve is then slowly closed by the action of the weight H acting against the resistance of the dashpot, and the pressure in chamber E gradually increases and closes the relief valve.

The capacity of a pressure regulator should be between 50 and 75 per cent of the maximum turbine discharge, depending on the type of plant and the conditions of operation.

From the nature of the case the pressure regulator is much better adapted for use under high heads and with small volumes of water than where the volumes to be handled are large.

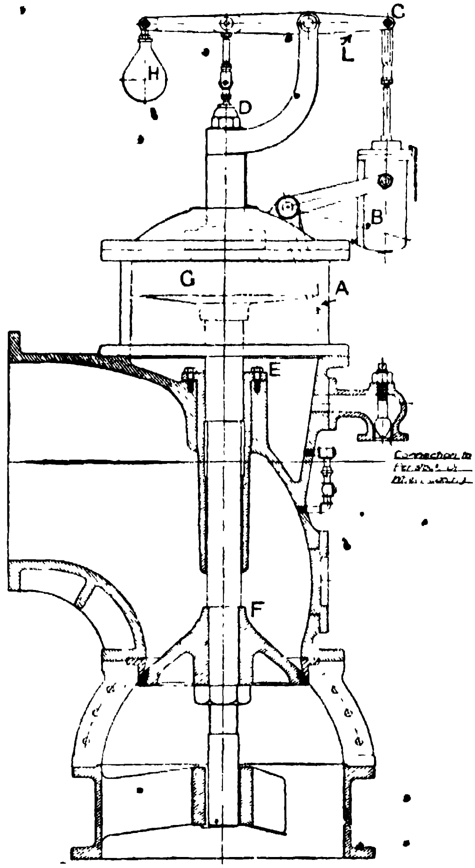


Fig. 135.—Governor-operated Pressure Regulator

100. Effect of Gates and Governor Connections.—Whatever the type of plant, a well-designed system of gates and gate connections is essential for close speed regulation. In all connections, simplicity, directness, and freedom from backlash are essential. The gates themselves should be as light, well-balanced, and frictionless as possible.

In enclosed Francis turbines fitted with wicket gates, the guide spindles are passed through stuffing boxes in the turbine casing, and the whole governing mechanism, with the exception of the gates and their pivot bearings, is removed from the action of the water. In turbines in open settings, the wicket gates, when fitted, are rotated either by means of an annular gear wheel—which gears with pinions mounted on the guide spindles, and which is rotated by means of a link coupled to an eccentric which receives its own motion from the relay mechanism—or a series of links mounted on the guide spindles are connected by levers to a central ring, which is rotated by means of the relay. In either case the gearing is submerged, and while accurate speed regulation is possible with either design, the submerged gearing needs to be designed on more substantial lines to compensate for its inaccessibility for examination.

101. Effects of Inertia of Rotating Parts.—During any sudden

change of load, there is a certain short period, whose duration depends on the design of the installation and on the sensitiveness and rapidity of action of the governor, during which the supply of energy to the turbine is greater or less than the demand, and in order to prevent the temporary fluctuation in speed during this period becoming excessive, some appreciable fly-wheel effect is necessary. This should be sufficient for the difference in the kinetic energy of the rotating parts, at the highest and lowest permissible speeds, to be equal to the maximum excess or defect of energy supply and demand during the period of changing load.

If the governor takes t seconds to complete the gate movement corresponding to a given change of load, and if the variation of head at the turbine during this period and the law of gate closure be known, the difference between the energy developed by the turbine and that absorbed at the turbine shaft during this period can be calculated. If this energy be represented by E, and if N be the mean speed and δN the fluctuation in speed in revolutions per minute, it may be shown that

$$\frac{\delta N}{N} = \frac{900gE}{\pi^2 I N^2} = \frac{2940E}{IN^2}.$$

For example, in the case of a turbine developing 5000 h.p. at 250 r.p.m., if 50 per cent load be thrown off instantaneously, and if the governor is capable of giving the required gate motion in 2 sec., the average horsepower generated by the turbine during this period, if the rate of change of input follows a straight-line law, will be $(5000 + 2500) \div 2 = 3750$ h.p. The output is 2500 h.p., so that E is the energy equivalent to 1250 h.p. for 2 sec., or $1250 \times 550 \times 2 = 1,370,000$ ft. lb. If under these conditions the increase in speed is not to exceed 5 per cent, so that $\delta N \div N = \cdot 05$, the necessary moment of inertia, I, of the rotating parts is equal to

$$\frac{2940 \times 1,370,000}{62,500 \times \cdot 05} = 1,290,000 \text{ ft. lb. units.}$$

Where no surge tank is fitted, or where any long length of pipe line is fitted between the surge tank and the turbines, it becomes necessary, in order to obtain an accurate estimate of the size of fly-wheel necessary to give governing within definite limits, to determine the pressure at each instant during the change of load, at the turbine gates or turbine nozzle. While this requires a somewhat tedious mathematical investigation, it may be done when the law of closure or of opening of the gates or nozzle is known.*

102. The Surge Tank.—The surge tank or stand pipe consists essentially of a vertical open pipe whose lower end is connected to the penstock as near to the turbines as possible, and whose upper end is above the surface-level in the supply reservoir (fig. 136 A). Any closure of the turbine gates is then accompanied by a flow up the stand pipe. The

* For this investigation the reader is referred to *Hydraulics*, Gibson (Constable & Co. 1912), pp. 222–42.

resultant retardation of the supply column and the rise in pressure, with a surge tank of adequate size, is greatly reduced.

The surge tank is, however, of greatest service on an increasing load, where, owing to the necessity for accelerating the supply column, the drop in pressure at the turbines would be excessive. In such a case any sudden

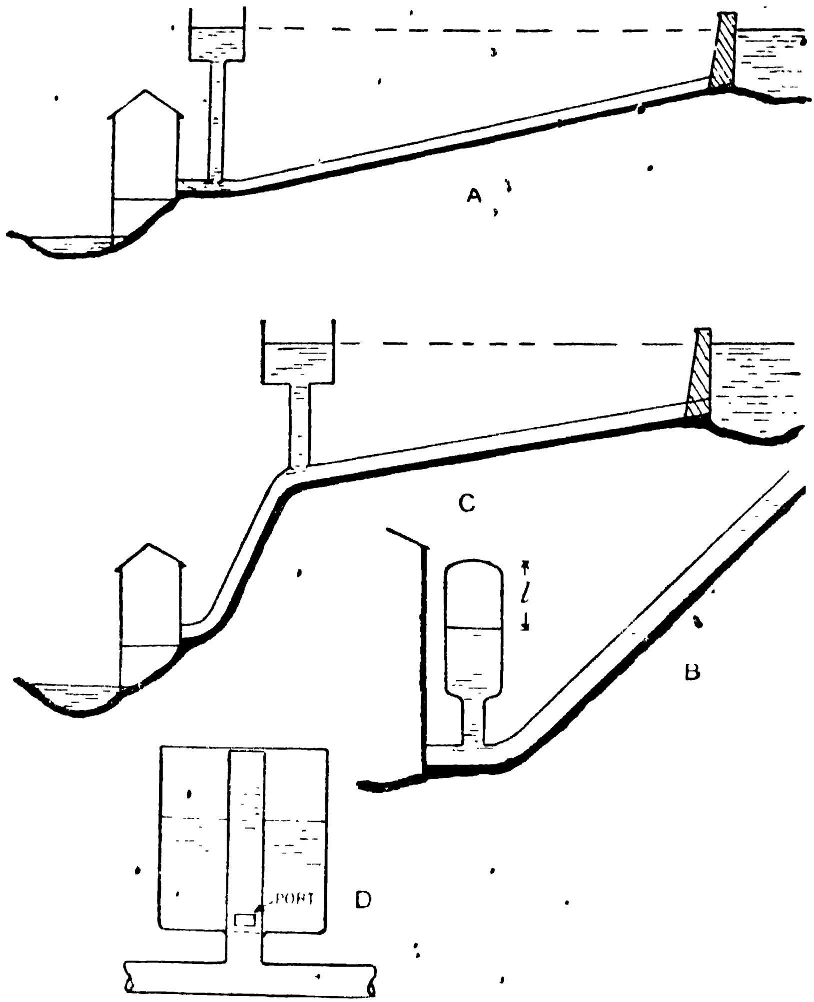


Fig. 136—Surge Tank

demand at the turbines is met by the flow down the surge tank, with a consequent reduction in the necessary acceleration in the penstock.

Where the head is so great as to render an open stand pipe impracticable, a closed stand pipe may be installed as shown in fig. 136 B. Compressed air is now supplied to the upper part of the tank by a small compressor of sufficient capacity to make up any leakage or absorption of air by the water.

Where there is high ground at some distance from the power house,

an open stand pipe may be installed there (fig. 136 C), and if the length of closed penstock between this and the turbines is too great for good regulation, a small auxiliary closed stand pipe may be installed adjacent to the turbines.

In the following investigation of the simple surge tank, let:

L = length of supply pipe line from forebay to surge tank;
R = ratio of sectional area of surge tank and main;
c = friction coefficient, so that friction loss in pipe line under velocity V = cV^2;
V_1 = velocity in pipe line under steady flow before change of load;
V_2 = velocity in pipe line under steady flow after change of load;
V = velocity in pipe line at any instant during unsteady period;
V′ = velocity in pipe between stand pipe and turbines at any instant during unsteady period;
y = height of water in stand pipe at any instant, above the level corresponding to steady flow with velocity V_1 (fig. 137).

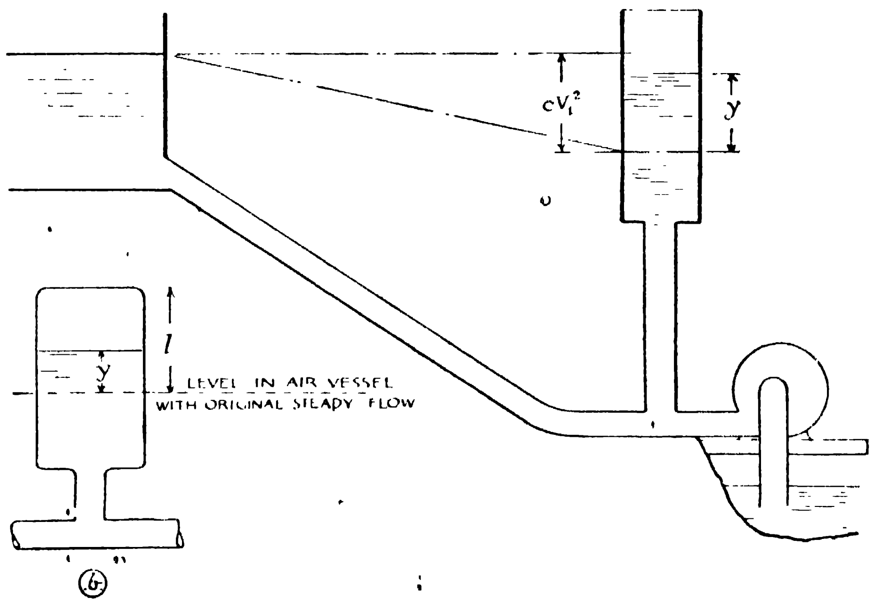


Fig. 137

Then at an instant t sec. after a given change of load, the head available for producing retardation in the pipe line is

$$y - c(V_1^2 - V^2),$$

and the equation of motion in the pipe line is

$$y - c(V_1^2 - V^2) = -\frac{L}{g}\frac{dV}{dt} \quad \ldots \quad (1)$$

Also, since the quantity of water entering the turbines equals the volume flowing down the pipe line, less that entering the tank,

$$R\frac{dv}{dt} = V - V' \qquad (2)$$

Multiplying (1) and (2) gives

$$[y - c(V_1^2 - V^2)]\,dy = -\frac{L}{gR}(V - V')\,dV \qquad (3)$$

$$\therefore \int y\,dy = \frac{L}{gR}\int(V' - V)\,dV + c\int(V_1^2 - V^2)\,dy \qquad (4)$$

At the point of maximum surge $V' = V_2$, and (4) becomes

$$\int_0^{y_m} y\,dy = \frac{L}{gR}\int_{V_1}^{V_2}(V_2 - V)\,dV + c\int_0^{y_m}(V_1^2 - V^2)\,dy, \qquad (5)$$

$$\text{or } y_m^2 = \frac{L}{gR}(V_2 - V_1)^2 + 2c\int_0^{y_m}(V_1^2 - V^2)\,dy \qquad (6)$$

The last term cannot be integrated mathematically unless V is known in terms of y. Since V is unknown, some approximate method is necessary to obtain a solution. Denoting the variable $c(V_1^2 - V^2)$ by z, it appears that when y is small, z increases slowly and dz is less than dy. As y approaches its maximum, z increases more rapidly than y, and dz becomes greater than dy. In view of this, R. J. Johnson * assumes that a sufficiently close approximation is attained by putting $dz = dy$ as an average over the whole summation, so that

$$c\int_0^{y_m}(V_1^2 - V^2)\,dy = \int_{z_1}^{z_2} z\,dz = \left[\frac{z^2}{2}\right]_{z_1}^{z_2}$$

$$= \frac{c^2}{2}\left[(V_1^2 - V^2)^2\right]_{V_1}^{V_2} = \frac{c^2}{2}(V_1^2 - V_2^2)^2,$$

leading to the final expression

$$y_m^2 = \frac{L}{gR}(V_1 - V_2)^2 + c^2(V_1^2 - V_2^2)^2 \qquad (7)$$

This expression holds for both acceleration and retardation.

Neglecting friction and any governor action, the motion in the tank would be simple harmonic, with a time T of complete oscillation, given by

$$T = 2\pi\sqrt{\frac{RL}{g}}\ \text{sec.}; \qquad (8)$$

while y_{max} would be equal to $\pm\sqrt{\frac{L}{gR}}(V_1 - V_2)$. $\qquad (9)$

* *Trans. Am. Soc. C. E.*, Vol. LXXVIII (1915), p. 760. Also 1908, p. 443.

Taking friction into account, the time of oscillation is increased. Johnson finds that the expression

$$T = \frac{\pi}{2}\sqrt{\frac{RL}{g} + c^2R^2(V_1 + V_2)^2} \text{ sec.} \ldots\ldots\ldots \quad (10)$$

gives a very close approximation to the actual time required to reach the first crest or trough of the wave.

After the first surge the oscillation is gradually damped out by the pipe friction, except where affected by the action of the governor (Art. 103). If the governor operates so as to close the gates during the upward surge, and to reopen them on the downward surge, keeping in step with the natural oscillation in the tank, the oscillations, instead of dying out, may, with a tank of comparatively small cross-sectional area, increase with time, in which case successful operation is impossible.

In deducing expression (7), it is assumed that the velocity of flow to the turbines at the instant of maximum surge is the same as V_2, the velocity of flow under steady flow when conditions have settled down after the change. Owing to the fact that the head at the turbines is not the same at the two instants, this will not, in general, be the case. Apart from this, expression (7), depending as it does on an approximation to the effect of friction, may be appreciably in error if the friction loss is relatively high. Thus, while the expression is very useful in making preliminary investigations, it should not be used for the determination of final designs. For such a purpose the method of arithmetical integration should be used. This method, while somewhat tedious, enables the effect of any known and variable factor to be taken into account.

Arithmetical Integration and Solution of Stand Pipe Surge.—Suppose the initial velocity to be V_1, and suppose a sudden alteration in the demand to V_2, which is to be constant after the initial change. If V_2 be less than V_1, the water will rise in the surge tank.

After a small interval of time, δt, let the velocity up the stand pipe be v_1. Then the mean upward velocity during this period will be very approximately $v_1/2$, and the height y above that obtaining before the change will be $v_1\delta t/2$ at the end of the period. At this instant the velocity in the main pipe line will be $V_2 + v_1R$, where R is the ratio of stand pipe area to pipe area, so that the mean acceleration dV/dt in the main during this period will be equal to

$$-\left\{\frac{V_1 - (V_2 + v_1R)}{\delta t}\right\}.$$

At the middle of the interval the height y will be $v_1\delta t/4$, and the velocity in the main will be $V_2 + v_1R/2$, so that the mean head available for producing retardation is

$$\frac{v_1}{4}\delta t - c\left\{V_1^2 - \left(V_2 + \frac{v_1R}{2}\right)^2\right\},$$

and, equation (1), p. 190, becomes

$$\left\{\frac{V_1 - (V_2 + v_1R)}{\delta t}\right\}\frac{L}{g} + c\left\{V_1^2 - \left(V_2 + \frac{v_1R}{2}\right)^2\right\} = \frac{v_1\delta t}{4}$$

from which, by substituting the known values of V_1, V_2, and δt, v_1 may be obtained.

Now taking a second period δt, and calling v_2 the velocity up the stand pipe at the end of this period, the mean velocity during the period is $(v_1 + v_2) \div 2$, and the height at the middle of the period will be equal to

$$y_1 + \frac{(v_1 + v_2)}{4}\delta t.$$

At this instant the velocity in the main is $V_2 + \frac{(v_1 + v_2)R}{2}$, and since $\frac{dV}{dt}$ during the period is $\frac{(v_2 - v_1)}{\delta t}R$, we have:—

$$\left(\frac{v_1 - v_2}{\delta t}\right)\frac{RL}{g} + c\left\{V_1^2 - \left(V_2 + \left(\frac{v_1 + v_2}{2}\right)R\right)^2\right\} = y_1 + \left(\frac{v_1 + v_2}{4}\right)\delta t.$$

This gives v_2, and since $y_2 = y_1 + \left(\frac{v_1 + v_2}{2}\right)\delta t$, it also gives y_2.

Similarly at the end of a third period, we have the equation

$$\left(\frac{v_2 - v_3}{\delta t}\right)\frac{RL}{g} + c\left\{V_1^2 - \left(V_2 + \frac{(v_2 + v_3)R}{2}\right)^2\right\} = y_2 + \left(\frac{v_2 + v_3}{4}\right)\delta t,$$

giving v_3, and so on.

If, for example, L = 500 ft., R = 8·0, c = ·03, V_1 = 4·77 f.s., V_2 = 1·94 f.s., and if an interval δt of 5 sec. be adopted, we have, at end of period (1):

$$\frac{(4\cdot77 - 1\cdot94 - 8v_1)500}{5 \times 32\cdot2} + \cdot03\{22\cdot75 - (1\cdot94 + 4v_1)^2\} = \frac{5v_1}{4},$$

which reduces to $v_1^2 + 55\cdot4v_1 - 19\cdot5 = 0$,

giving $v_1 = \cdot352$ f.s.

$\therefore y_1 = \cdot352 \times 2\cdot5 = 0\cdot88$ ft.

and $V = 1\cdot94 + (8 \times \cdot352) = 4\cdot786$ f.s.

At end of period (2):

$$\frac{(\cdot352 - v_2)\,500 \times 8}{5 \times 32\cdot2} + \cdot03\{22\cdot75 - (1\cdot94 + 4(\cdot352 + v_2))^2\}$$

$$= 0\cdot88 + \left(\frac{\cdot352 + v_2}{4}\right)5,$$

reducing to $v_2^2 + 56\cdot1 - 16\cdot2 = 0$,

giving $v_2 = \cdot289$ f.s.

$\therefore y_2 = \cdot88 + \frac{(\cdot352 + \cdot289)5}{2} = 2\cdot48$ ft.

and $V = 1\cdot94 + (8 \times \cdot289) = 4\cdot26$ f.s.

Proceeding from step to step in this way, such curves as fig. 138 are obtained. An upward surge of 4·2 ft. is followed by a downward surge of 2·8 ft. below the original level in the surge tank, which would be followed by a smaller upward surge, and so on until the level settles down to that obtaining under the new steady conditions. The time to reach the crest of the first surge is approximately 20 sec.

Calculating the height of the first surge and the time to reach the maximum height by formulæ (7) and (10) gives a height of 3·98 ft. and a time of 17·7 sec.

In investigating a surge-tank problem to fix the final design, the intervals of time should be taken more closely than in this example. Usually eight or ten intervals are advisable up to time of the first surge.

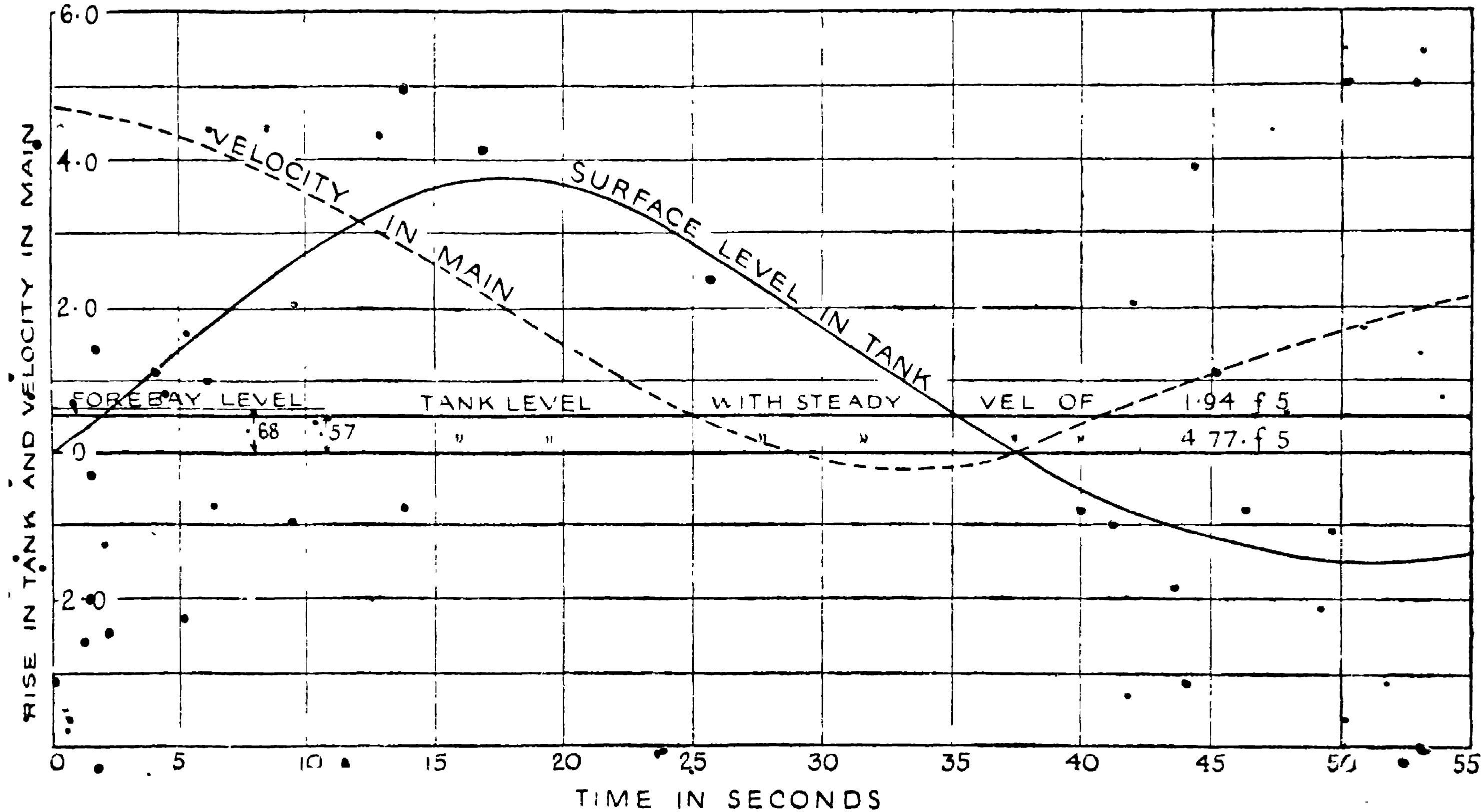
103. Effect of Governor Action on Surge. In the preceding example it was assumed that the velocity in the pipe leading from the surge tank junction to the turbines adopted a definite value after the change, and maintained this value during the succeeding changes of level in the surge tank. Actually, however, if the length of pipe between tank and turbines is small, the head at the turbines will vary sensibly as the level in the tank, and since the quantity of water required to maintain any definite load is, within narrow limits, inversely proportional to the head if the turbine efficiency is constant, it follows that if the governor is sufficiently sensitive, and if the surge period is fairly long, the velocity of flow in this pipe will not be constant, but will vary inversely as the head measured between the tail race and the level in the surge tank. This has the effect of increasing the magnitude of the surge following any change of load, and may, if the governor keeps in step with the surges, in some cases cause these to increase instead of diminish with time, in which case satisfactory operation is impossible.

The introduction of a mathematical term representing this effect into the equations of motion leads to an expression incapable of any simple solution. This effect can, however, readily be investigated by the method of arithmetical integration.

In the last example, for instance, if the head measured from forebay level is 50 ft., the working head measured from the level in the tank will be 49·32 ft. before the change of load, and 49·89 ft. when the flow has settled down to 1·94 f.s. after the change. The power developed will then be 41 per cent of the initial output. Now suppose that the governor is capable of keeping the speed constant, and that, in consequence, the demand at any instant is universely proportional to the head, instead of being constant. The mean head during period (1) is approximately 50·1 ft., the flow required to give the reduced output will be

$$1·94 \times 49·89 \div 50·1 = 1·93 \text{ f.s.}$$

This value is so near the assumed value of 1·94 that any alteration in v_1 or y_1 may be neglected in this case. During period (2), the mean head



ig 138 —Fluctuations of Velocity and Pressure in Pipe Line fitted with simple Surge Tank

as obtained from the curve is 51·5 ft., and the necessary value of V_2 during this period is

$$1{\cdot}94 \times 49{\cdot}89 \div 51{\cdot}5 = 1{\cdot}88 \text{ f.s.}$$

Using this value instead of 1·94 in the calculations relating to period (2) on p. 193 gives new values of y_2 and v_2. On sketching in the new curve it will be seen whether the assumed mean head during the period is sufficiently near the actual mean. If not, a second and closer approximation can readily be made. In this way the whole of the cycle can be investigated, and the maximum fluctuation of level corresponding to any given change of load, both increasing and diminishing, can be obtained for a stand pipe of any given area.

In practice the stand pipe is to be designed to keep the fluctuation of level to within a certain percentage of the working head, under a given sudden diminution of load. While much depends upon the special requirements of the plant, the total fluctuation in head should not in general exceed about 5 per cent under a sudden alteration of 20 per cent in the load. If possible without an undue expenditure in surge-tank construction, the percentage fluctuation in head should be reduced to one-half the above amount. Inserting appropriate values of y_m, L, V_1 and V_2 in equation (7) gives a value of R, the ratio of stand-pipe area to conduit area. The actual value required will in general be somewhat greater than this calculated value, and a value 10 per cent in excess of this may be used as the basis of a more detailed examination by arithmetical integration.

If this shows the fluctuation still to be too great, a second approximation can usually be made which gives the required fluctuation within narrow limits, remembering that when near the correct size the fluctuation is approximately proportional to $\sqrt{R}$.

104. Closed Stand Pipe.—In the case of a closed stand pipe let l be the length of the tank above the original water-level (fig. 137*b*) and p the original air pressure in feet of water. Then equation (1), p. 190, becomes:

$$\frac{pl}{l-y} + y - c(V_1^2 - V^2) = \frac{L}{g}\frac{dV}{dt},$$

while equation (7) becomes:

$$y_m^2 = 2pl \log_e\left(\frac{l - y_m}{l}\right) + \frac{L}{gR}(V_1 - V_2)^2 + c^2(V_1^2 - V_2)^2.$$

This expression holds for both acceleration and retardation by giving the correct sign to y.

From the form of this expression it is evident that a given volume of air chamber is most effective when the sectional area is as large and the length l as small as possible.

In this case also the entire cycle of a surge may readily be investigated by arithmetical investigation.

The closed stand pipe suffers from the disadvantage that an air com-

pressor requires to be installed to maintain the requisite volume of air above the water surface. Unless very carefully designed, to suit the characteristics of the particular installation, sympathetic surges are apt to be set up, which render successful regulation impossible.

105. The Differential Surge Tank.—With a view to preventing any augmented oscillations in the surge tank due to governor action or to synchronism of loads, R. D. Johnson * has suggested the use of the differential surge tank.

This consists of a simple stand pipe or riser freely connected to the conduit, and supplemented by a larger tank which surrounds the riser and communicates with it through one or more openings of restricted size near the bottom of the tank (fig. 136 D). The diameter of the riser is usually from ·75 to 1·0 times that of the conduit.

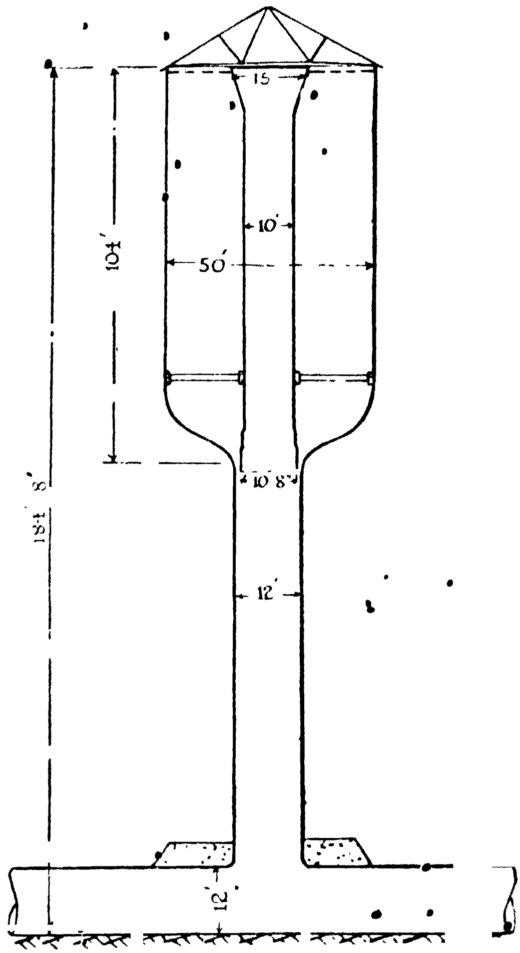


Fig. 139.—Differential Surge Tank of Salmon River Power Co.

Fig. 139 shows a steel differential surge tank installed at the plant of the Salmon River Power Co., Altmar, New York.† This is supported by ten steel columns, raising the top of the tank 185 ft. above the foundations. This tank is located on the top of a hill, and is connected to the lower end of an 11-ft. pipe line 9500 ft. long, between the dam and the tank. This pipe line terminates in a 12-ft. distributor pipe, from which four 8-ft. penstocks, 380 ft. long, convey water to the power plant below. The entire plant is designed for 40,000 h.p. A 12-ft. tee on the distributor connects to the 12-ft. riser pipe of the surge tank. The portion of the riser in the tank is 10 ft. in diameter, and is carried by the walls of the tank. This pipe is enlarged to 10 ft. 8 in. where it

* *Trans. American Soc. C. E.*, Vol. 78 (1915), p. 760. Also *Trans. Am. Soc. C. E.*, 1908.
† *Engineering News*, 16th July, 1914, p. 150.

enters the 12-ft. riser, forming an annular opening or port between the two risers. Freezing in winter is prevented by the circulation of hot air around the top of the tank, from a heater at the foot of the riser.

In the case of a sudden change of load the level in the riser alters quickly, allowing the acceleration or retardation in the conduit to take place more rapidly than with a simple surge tank. The difference of level between the water in the riser and the tank then causes a flow through the connecting ports, which gradually equalizes the levels. In the correctly designed tank the area of the tank and port are so arranged as to enable this to take place just as the velocity in the conduit attains its final steady value.

It is evident that since the initial pressure change at the turbines is greater than with the simple surge tank, extra work will be thrown on the governors, and the inertia of the rotating parts will require to be greater in order to maintain the same temporary fluctuation in speed. Actually, however, the modern governor is capable of dealing satisfactorily with such changes as may occur in any normal plant.

In order to deduce mathematical expressions for the changes in level and velocity which take place with such a surge tank, certain approximate assumptions require to be made.

These are:

1. The area of the riser is considered to be so small that its variation in water-level takes place within an interval of time which is negligible compared with the time to produce complete acceleration of the water column in the conduit.

2. The port area is assumed to vary in size, so that when a sudden change of level takes place in the riser, this water-level remains stationary at the new position, until complete acceleration is effected in the conduit.

3. The inertia and friction in the riser and tank are neglected.

While these assumptions are only approximate, experience shows that the results obtained by their use are in very close agreement with experiment in any normal case.

Adopting the same notation as for the simple surge tank, with the following additions:

A = area of conduit,
F = area of tank in excess of area of riser,
a = area of port, assuming unity as the coefficient of discharge,
$Z = \sqrt{\frac{y_1}{c} + V_1^2}$,
y_1 = initial sudden change of level in riser (assumed instantaneous),

we have, during acceleration:

$$\frac{L}{g}\frac{dV}{dt} = y_1 - c(V^2 - V_1^2) \quad \text{. . . . . (1)}$$

from which, by integrating between the limits V_1 and V, we get for the time t, to attain any given velocity V,

$$t = \frac{L}{2gcZ} \log_e \frac{(Z - V_1)(Z + V)}{(Z + V_1)(Z - V)} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

Multiplying both sides of (1) by AV, we have

$$A\int_0^t V\,dt = \frac{AL}{g}\int_{V_1}^{V} \frac{V\,dV}{y_1 - c(V^2 - V_1^2)},$$

which gives the volume flowing through the conduit in time t. Since the volume entering the turbines during this period is, by hypothesis, AV_2t, the volume taken from the tank will be

$$AV_2t - \frac{AL}{g}\int_{V_1}^{V} \frac{V\,dV}{y - c(V^2 - V_1^2)} = yF.$$

Integrating and simplifying, and substituting for t from (2) gives

$$y = \frac{AL}{2gcF}\left\{\frac{V_2}{Z}\log_e \frac{(Z - V_1)(Z + V)}{(Z + V_1)(Z - V)} - \log_e \frac{Z^2 - V_1^2}{Z^2 - V^2}\right\}, \ldots \quad (3)$$

giving the relationship between the water-level in the tank and the corresponding velocity in the conduit.*

When $V = V_2$, $y = y_1$ if F is the current area, so that

$$F = \frac{AL}{2gcy_1}\left\{\frac{V_2}{Z}\log_e \frac{(Z - V_1)(Z + V_2)}{(Z + V_1)(Z - V_2)} - \log_e \frac{Z^2 - V_1^2}{Z^2 - V_2^2}\right\} \ldots \quad (4)$$

The head producing flow through the ports at any instant is $y_1 - y$. Calling this h_p, and substituting the value of y_1 and y from (4) and (3), gives

$$h_p = \frac{AL}{2gcF}\left\{\frac{V_2}{Z}\log_e \frac{(Z - V)(Z + V_2)}{(Z + V)(Z - V_2)} - \log_e \frac{Z^2 - V^2}{Z^2 - V_2^2}\right\}.$$

The necessary flow through the ports is $A(V_2 - V)$ f.s., so that

$$a\sqrt{2gh_p} = A(V_2 - V);$$

$$\text{and } a = \frac{(V_2 - V)\sqrt{A}}{\sqrt{\dfrac{L}{CF}\left\{\dfrac{V_2}{Z}\log_e \dfrac{(Z - V)(Z + V_2)}{(Z + V)(Z - V_2)} - \log_e \dfrac{Z^2 - V^2}{Z^2 - V_2^2}\right\}}}. \quad (5)$$

When $V = V_1$, at the beginning of the cycle,

$$a_o = \frac{A(V_2 - V_1)}{\sqrt{2gy_1}}; \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

* This treatment is that of Mr. Johnson. *Trans. Am. Soc. C. E.*, Vol. LXXVIII (1915), p. 770.

while when $V = V_2$, the expression becomes indeterminate, but approximates to

$$a_1 = \left\{\frac{AFy_1}{L}\left(1 - \frac{c(V_2^2 - V_1^2)}{y_1}\right)\right\}^{\frac{1}{2}}, \qquad (7)$$

which is less than a_0.

In practice a port of constant area is used. If designed from equation (6) this will be somewhat too large for the best results, although the error

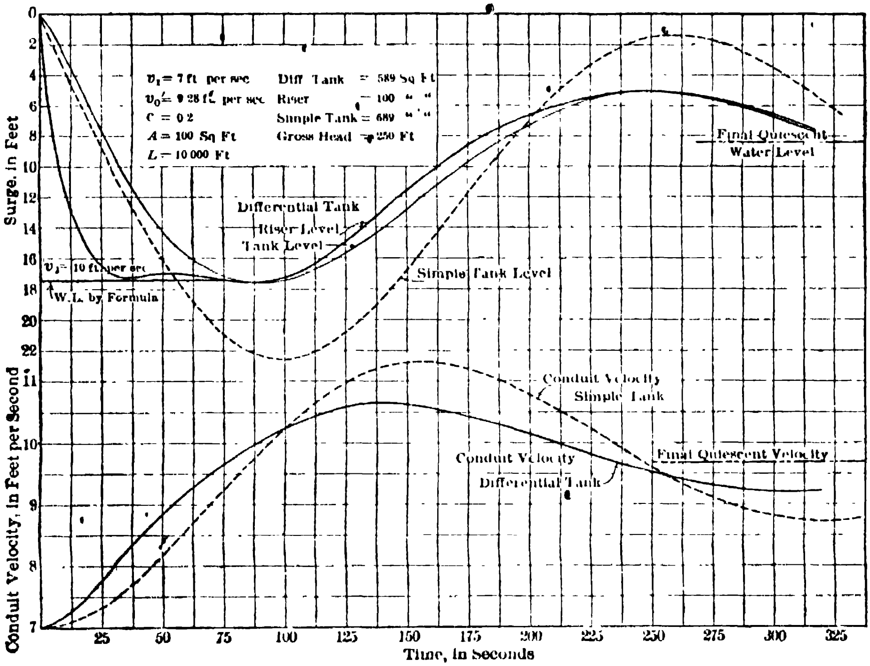


Fig. 140

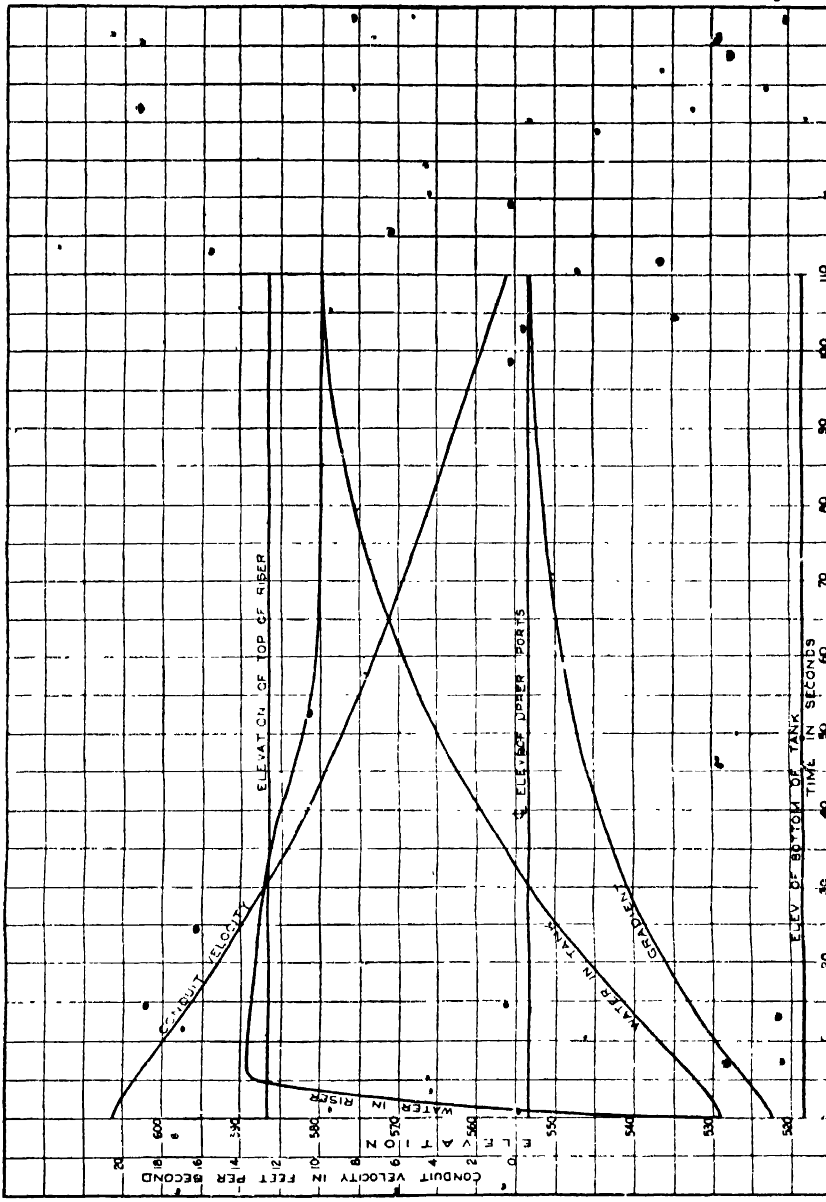
would not, in general, be serious. Its best area is preferably fixed by trial, using the method of arithmetical integration as outlined for the simple surge tank. In every case, owing to the approximations made in the mathematical analysis, the results should be checked, for the purpose of the final design, by the arithmetical method.

106. Retardation.—Having designed the tank of ample diameter and depth for the greatest increase of load for which good regulation is required, it is necessary to fix its height in order that it may store all the water overflowing from the top of the riser on a sudden shut-down.

The height d, measured above forebay level, to which the riser and tank must be carried for this purpose, is given by

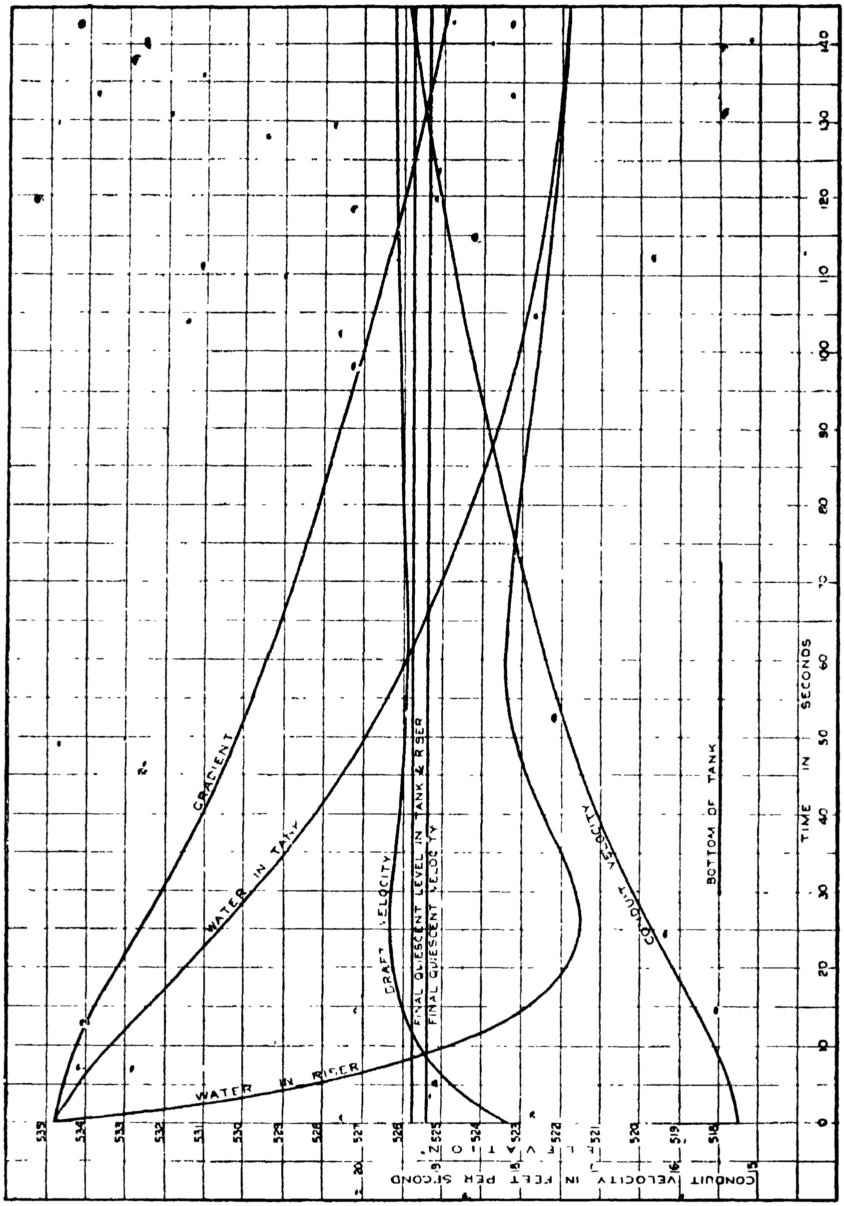
$$d = y_1 + cV_2^2.$$

It should be borne in mind that a larger value of V_2 than that corresponding to the maximum steady load demand may occur at the instant of shut-down,



which may happen to coincide with that of maximum conduit velocity during the second quarter cycle following an increase of load. The critical velocity producing a maximum rise may be shown to be equal to

$$\frac{1}{10}\sqrt{\frac{AL}{F}} \text{ ft. per second.}$$



and the extreme necessary value of d for the worst possible conditions, to be equal to

$$\frac{AL}{175cF}\ \text{ft.*}$$

* For proofs of these expressions and for a very detailed investigation of the whole problem, Mr. Johnson's paper before the *Am. Soc. C. E.* should be consulted.

The curves of fig. 140, which are taken from Mr. Johnson's paper, show, in a typical case, the difference between the behaviour of a simple surge tank and a differential tank on an increasing load.

The curves of figs. 141 and 142 show the effect of a differential tank, respectively on a complete shut-down, and when the load is suddenly changed from 80 per cent to full load.* The tank, which is installed on No. 3 conduit of the Ontario Power Co., Niagara, is 60 ft. in diameter and 94 ft. high, and is connected to the conduit, which is 13·5 ft. diameter, by 77 ft. of pipe of the same diameter. The conduit is 6882 ft. long.

Fluctuations of Speed in Practice. As already indicated, the most perfect governor is incapable of giving close speed regulation if the hydraulic conditions of the installation are essentially bad. With reasonably good hydraulic conditions, and without any special fly-wheel, a good modern governor will enable the speed of a reaction turbine to be maintained within about

2·5	per cent	for a	25	per cent	load change,
5–6	,,	,,	50	,,	,,
12–15	,,	,,	100	,,	,,

These are the instantaneous fluctuations of speed. The final fluctuation in speed may in general be kept within

1	per cent	for a	25	per cent	load change,
2	,,	,,	50	,,	,,
4	,,	,,	100	,,	,,

In a Pelton wheel installation with a jet deflector and governor-operated needle the fluctuations of speed are greater on an increasing than on a diminishing load, owing to the necessity for the acceleration of the supply column on an increasing load. The following shows typical values of the instantaneous fluctuation in speed in such an installation:

12–15	per cent	for a	100	per cent	load change	(load on).
4–5	,,	,,	100	,,	,,	(load off).
5–7	,,	,,	50	,,	,,	(load on).
3–4	,,	,,	50	,,	,,	(load off).
3–4	,,	,,	25	,,	,,	(load on).
2–3	,,	,,	25	,,	,,	(load off).

The final fluctuations are sensibly the same as in a reaction turbine.

* Report of the H.-E. Commission of Ontario, 1918, Vol. I.

CHAPTER X

General Arrangement of Stations

107. The preceding chapters have given a general idea of the various components of the hydraulic portion of hydro-electric schemes. The object of this chapter is to discuss the appropriate methods of developing the power available by means of this equipment. It is impossible to lay down absolute rules, as the local conditions and variations in cost of labour and raw materials as well as of the machinery will always influence the scheme largely.

It cannot be too strongly insisted upon that the development must be planned with future possibilities always in mind. Experience has repeatedly shown that the demand for power has increased much more rapidly than anticipated, and original plans not permitting of full development have hampered extensions and injured the financial prospects very greatly.

Chapter II outlines the methods of ascertaining the flow of water to be relied upon as the basis of the scheme. Optimism in this regard is to be deprecated, and the figure finally adopted should be very conservative unless records extending over a long series of years are available. At the same time the scheme must be so arranged that extra plant can be installed if and when the flow is proved to be greater than the basic figure.

108. The head which can most conveniently be developed has next to be settled. This is often determined by the physical conditions, but if the gradient is fairly uniform there may be a number of possible locations for both the intake and the power house, while the possibility of increasing the head by increasing the height of the dam also requires careful consideration. The only criterion is that of cost as compared with the amount of power which can be developed and sold, and no general rules can be formulated. A sufficient head must be created to give the power required with the quantity of water available, and a wide margin to allow for future development in demand is highly desirable. For low-head schemes this necessity of creating a sufficient head may be the deciding factor in settling the height of the dam. For high-head schemes the question becomes one of storage or of raising the water to a sufficient level for it to be drawn off conveniently. The possible sites for the dam have to be examined with these objects in view.

109. From the dam the water is conveyed in a channel or pipe line of one of the types described in Chapter VII. For low-head schemes, owing to the large volume of water, an open canal is usually necessary, but it is subject to the disadvantage that if the level of the head water may vary appreciably from time to time, the canal must have a sufficient depth to enable the full head to be utilized, and at the same time to give an adequate

water way when the water is at its lowest level. A further disadvantage is that open canals usually entail higher maintenance and protection costs, as debris may be washed into them from the slopes above.

If a low-pressure pipe is used there is much more freedom of choice as to its location. For schemes with storage, the advantages of using closed pipes instead of canals are even more pronounced. Open canals must be located at levels corresponding with the lowest level to which the storage reservoir is to be drawn down, and this usually entails the loss of any extra head above this level at times of high water. In such cases the head gate controlling the flow into the canal must be adjusted so as to allow sufficient water to pass at all times, and a reduction in the demand for power will involve the wasting of the excess water. If a canal is used, some subsidiary storage or pondage should be arranged, if at all possible, at the head of the pipe lines to hold such excess water until it is required to meet a heavy demand. If this be done, the wastage can be considerably reduced. If a low-pressure pipe is used, a surge tank or open forebay will probably be necessary for its protection against water-hammer action, while the variations in level, corresponding to similar variations in the main reservoir, must be taken into account in determining the height of the walls in the forebay.

It is desirable to keep any pressure pipe lines as short as possible. In the first place they usually form a very expensive item of the equipment, and a long pipe line involves difficulties owing to water-hammer action. As a rough rule it may be taken that the length of these pipes should not be more than five or six times the effective head, otherwise special devices will be necessary for dealing with the surges. For this reason, whenever practicable, the water should be led to a point on the hillside immediately above the power station before entering the high-pressure pipe line, and at this point there should be a subsidiary storage reservoir if the water is brought by an open canal, or a surge tank if by a low-pressure pipe.

As already pointed out in Chapter VII, the material, number, and diameter of the pipe lines can only be settled after trial calculations of the costs of the various alternatives. The question as to whether an inter-connecting pipe is desirable depends on the value of the additional degree of reliability obtained, compared with the additional cost. For large installations the tendency is towards keeping the turbine and its corresponding pipe line an isolated unit.

110. In connection with the power house itself, the number and type of turbines to be employed has first to be decided. The broad principles are that the units should be as large as possible, and at the same time that the capital locked up in the stand-by units should not be excessive. Remembering that the cost per kilowatt is reduced as the sizes of the turbine and generator are increased, it is obvious that for the same shaft horse-power in the working units it will often be advantageous to instal fewer and larger units, even though this involves a larger unit standing idle as

a spare. Furthermore, the present tendency is towards still larger units, so that for the initial installation the size should be stretched to the utmost. In an independent station of moderate size, three working units with a fourth as spare, or four working units with a fifth as spare may be taken as generally appropriate for the completed station, even if this means putting in one unit for the whole of the initial load with a second as stand-by.

Where a number of stations are interconnected, feeding into a common transmission line, the number of units may be reduced to two, or even one in each station. In this case, as the load demand is reduced one or more stations are cut out of service, while each station serves as a stand-by to the remainder.

There are naturally other limitations to be borne in mind. For low-head schemes the number of units may have to be increased considerably. In the case of the single-wheel vertical type of turbine, the difficulties of manufacturing and transporting wheels of very large diameter may be sufficient to determine the maximum practicable size, or it may be that the dam, which is in any case necessary, is long enough to accommodate the requisite number of small units, whereas larger ones would entail widening the dam in order to provide the necessary foundation room and efficient settings

For high-head schemes the quantity of water that may be conveyed in a single pipe is limited, since there is an economical limit to the diameter for a given head, and the unit may be restricted in capacity by this feature. The question of transport of the various parts has to be considered, and the generator rotor itself will often provide a limit in this direction. Briefly, the largest possible units should be selected, but only after a complete and careful study of the controlling factors.

The selection of the type of turbine has next to be made. The limitations imposed by the speed of the generator and by the specific speeds of the various types of turbine have been discussed in Art. 95.

For low heads, especially if the water is led to the power station in an open canal, open type turbines have the advantage of being simple and cheap. It must, however, be remembered that the cost of excavation of the turbine pit and the necessary concrete must be set against the saving in cost due to the omission of a turbine casing, and naturally as the head and therefore the depth of the turbine pit increases, this advantage tends to disappear. For small turbines the economical limit in this direction may be 25 ft., but for very large machines, say 15,000 h.p., it may be as much as 70 ft.

Fig. 143 shows what is probably the simplest possible arrangement of a single-wheel vertical unit. No governor is shown. For small powers the drive may be transmitted to the generator by a quarter-turn belt, and for larger powers by bevel gearing; or the generator may be arranged for direct coupling, with an overhead thrust bearing to carry the weight of the rotating parts and the hydraulic thrust on the turbine runner. This type of setting has the disadvantage that the guide vane mechanism is

submerged, and cannot be inspected or repaired without draining the wheel pits.

Fig. 144 shows a simple open-setting arrangement of a single-wheel horizontal shaft unit, with direct coupling.

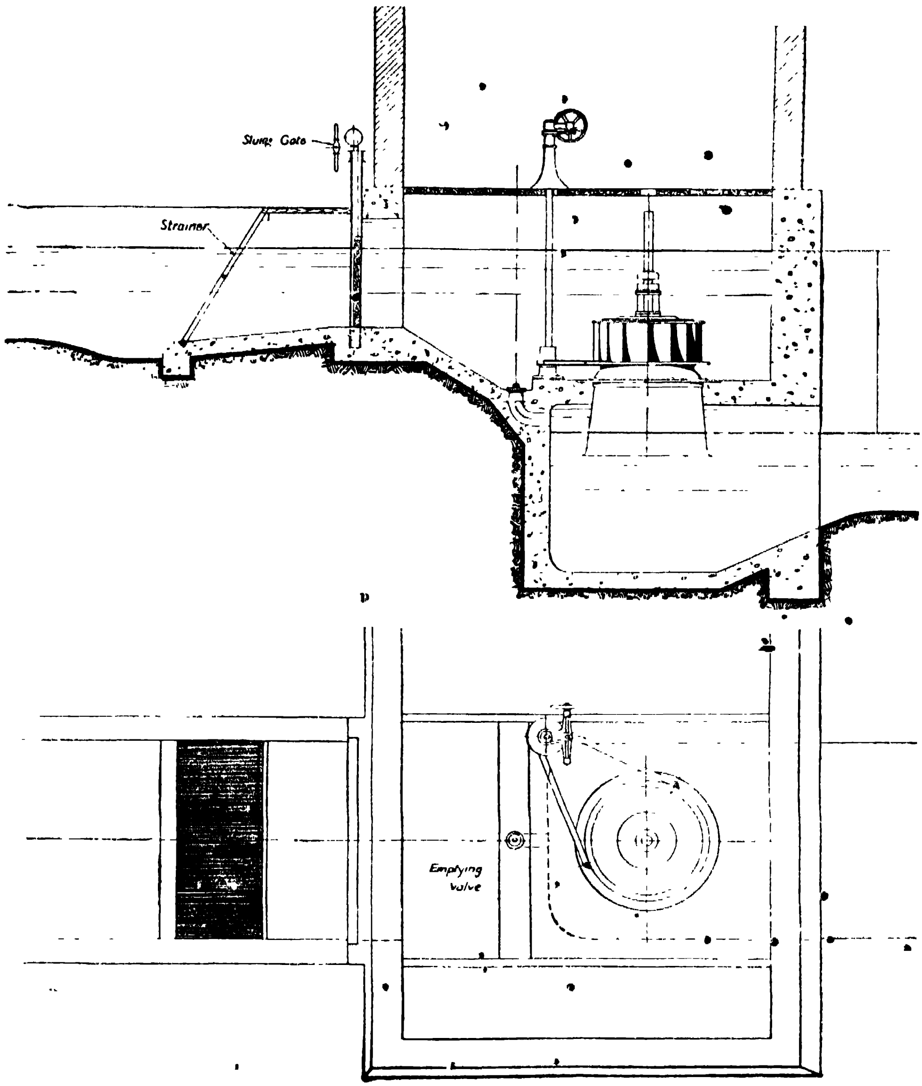


Fig 143 —Arrangement of Single Vertical Open Turbine

Vertical units offer many advantages over horizontal ones, especially if there are great variations in the tail race level. A horizontal arrangement necessitates keeping the station floor above the highest tail-water level or alternatively making the lower part of the machine room watertight.

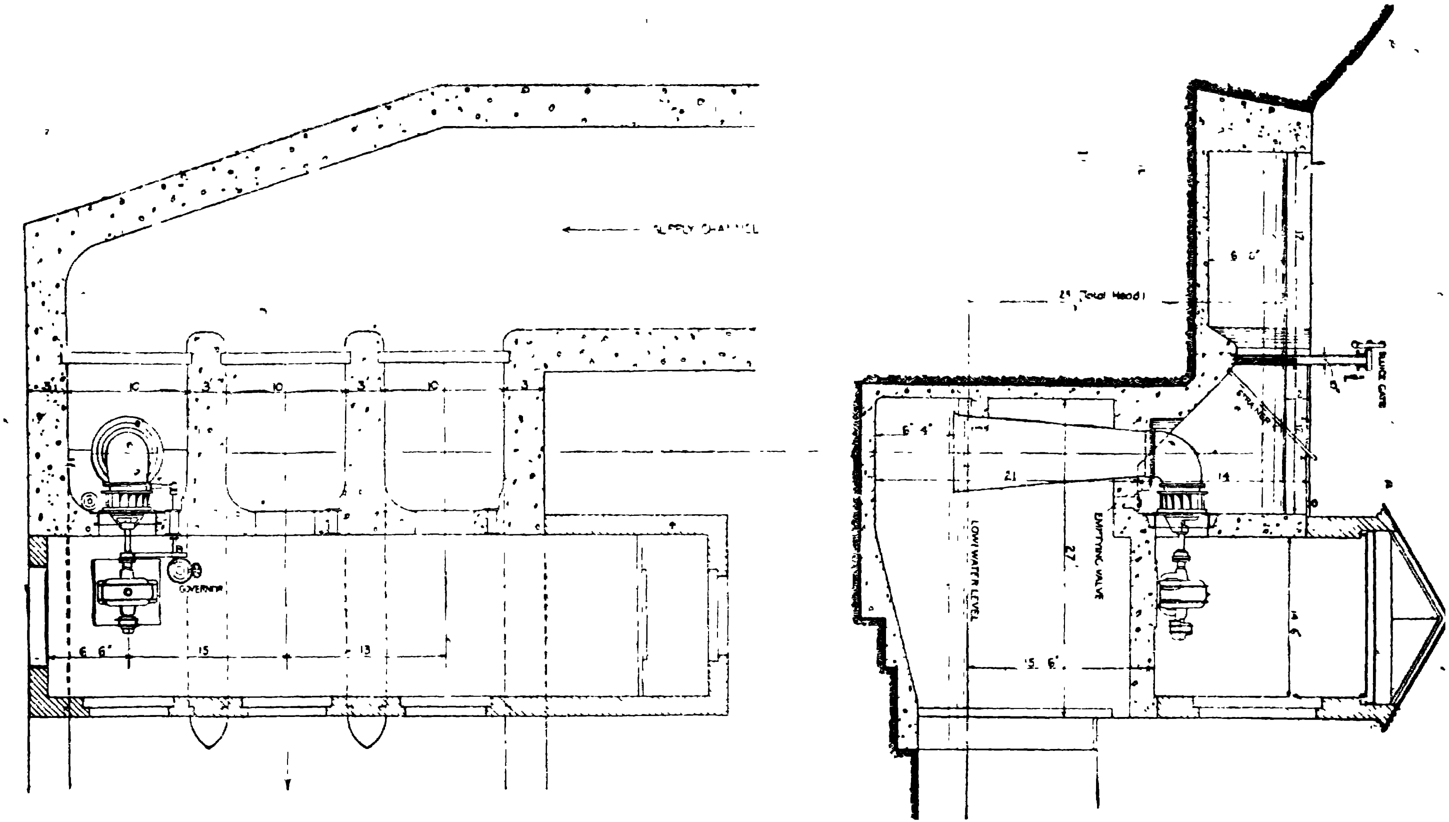


Fig. 144.—Single-wheel, Direct-coupled, Horizontal Shaft Units

Again, with a given depth of forebay, a greater depth of immersion of the wheel may be given with a vertical shaft unit. Where it is necessary to install a low-head turbine at a considerable distance from the reservoir, and at the same time at some height above tail-race level, the head of water above the turbine is of necessity small, and a sudden increase in load may reduce the level to such an extent as to draw air into the wheel, destroying the vacuum in the draft tube and stopping the turbine. To prevent this a minimum depth of 3·5 to 4 ft. should be allowed above the turbine, and the forebay must be of ample area. The necessity for complying with this condition, and at the same time for placing the station floor at a sufficiently high level, may render the horizontal shaft arrangement impossible, and may necessitate a vertical turbine with its shaft carried beyond the level of the highest floods.

One disadvantage of the vertical shaft arrangement is that it does not lend itself to the grouping of two or more runners on one shaft, since such an arrangement requires very complicated foundations, renders the parts inaccessible, and greatly increases the vertical height. Where a multiple-wheel unit is required the horizontal shaft arrangement is almost always adopted. In many of the earlier low-head installations, horizontal units were used with as many as four or even eight runners on the shaft. Owing to the many disadvantages of this type of construction (Art. 91), and the fact that improved design has enabled much higher speeds of rotation to be obtained from a single wheel than were possible in the past, turbines having more than two runners are now seldom constructed. The double horizontal machine is still often installed, especially where the question of space is not of great importance, and where the head of water above the turbine is ample to prevent ingress of air.

In such a turbine the bearing at one end of the shaft may be submerged. With clean water a lignum vitae bearing gives satisfactory service under such conditions, but if the water carries much sand or grit, submerged bearings are very unsatisfactory. In such a case the type of installation shown in fig. 145 gets over the difficulty. Here an inspection tunnel is provided for the back bearing.

One drawback to the use of the vertical single-wheel type for small units is due to the fact that vertical shafts are unusual in ordinary electrical engineering practice, and that small vertical generators are not standard and are correspondingly dear. Also there is still a certain amount of distrust of thrust bearings, in spite of the fact that they have proved themselves quite reliable in practice. The former of these difficulties can be surmounted by using a bevel gear in connection with a horizontal shaft generator. Incidentally this enables a higher rotational speed to be obtained and a smaller and cheaper generator to be used, and in some cases enables two or more turbines to be coupled to a single generator.

For large units the generators have in any case to be specially designed, while the thrust bearings which have been developed for these large machines have proved very reasonably reliable. The foregoing dis-

advantages thus disappear, and in view of the many advantages, modern practice has tended in the direction of single-wheel vertical units for both low and medium heads.

In spite of this, the choice between a vertical and a horizontal unit is only to be made after a detailed examination of the particular scheme. Comparing a single-wheel vertical unit in a concrete scroll setting and a double-wheel horizontal turbine in an open pit, the single wheel will be larger and the setting more efficient, so that the hydraulic efficiency will be higher. This, however, is partly counterbalanced by the fact that since the speed of the single wheel is lower its generator will be larger, and, running at the lower speed, will in general have a lower efficiency. In many cases the cost of the concrete moulded volute chamber is so great as appreciably to counterbalance the slight gain in efficiency, and there are some grounds for believing that this vertical arrangement, which has been so much used during the last few years, will not be adopted so frequently in the future. These remarks, of course, do not apply to cases where the variations in the water-level are large, for which cases the vertical arrangement has special advantages.

For heads greater than are suitable for an open setting, the turbine must be enclosed in a casing and be supplied with pressure water through a pipe line. The point at which this becomes necessary depends to some extent on the size of unit and also on the topography of the site. If the ground slopes gradually for some distance above the power station, the cost of a canal of sufficient depth may be prohibitive, and a pipe line becomes necessary. Even in this case an open setting is sometimes used, with the pipe feeding a forebay whose walls are carried to a sufficiently high level, and which acts as a surge chamber. With a very low head and a long canal, the danger of drawing air into the turbine when the load is increased, owing to an insufficient depth of submersion, may necessitate the turbine being cased, and fed from an open forebay through a short pipe line.

The disadvantage inherent in the open setting for a vertical shaft turbine, due to the fact that the guide-vane mechanism is submerged and cannot be inspected and repaired without draining the wheel pit, has led to the use of cased turbines in many recent important installations, even where the head is suitable for an open setting. In such a case the guide-vane ring is usually surrounded by a spiral volute chamber into which the pressure water is admitted from the forebay, and from which it is delivered with uniform velocity around the periphery of the guide ring. For heads up to about 80 ft. and for large units, modern practice favours the moulding of the volute chamber in the concrete of the substructure, reinforcement being added if necessary to provide sufficient strength.

Fig. 146 shows such a unit designed to give 8000 h.p. at 83 r.p.m. under a head of 40 ft. It will be noted that the power station is narrower than that of a horizontal shaft installation. The foundations are reduced in bulk, but owing to the more expensive forms required, the cost of the

foundation is not necessarily less, and may indeed be more than that for a horizontal shaft unit. The power station is, however, more easily located

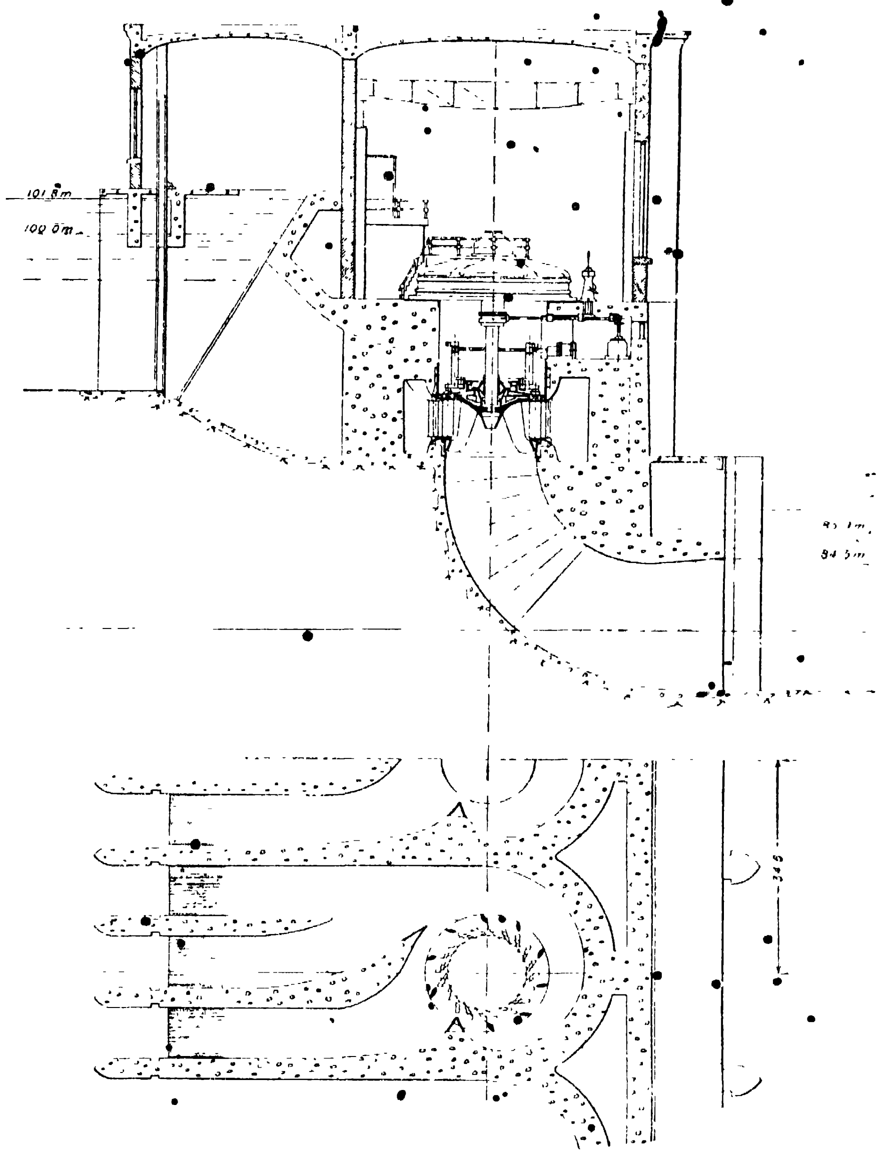


Fig 146

if the space available is limited, and if the station forms part of the structure of a dam, the advantage is obvious.

Fig. 147 shows an example of a fairly recent plant installed at Olten Gosgen, Switzerland.* This unit gives 10,240 h.p. at 83·3 r.p.m. under a head of 55 ft. The supporting of the generator on pillars so as to do away with an intermediate floor is noteworthy as reducing the cost of the foundations, and making the moving parts more readily accessible from the main-floor level.

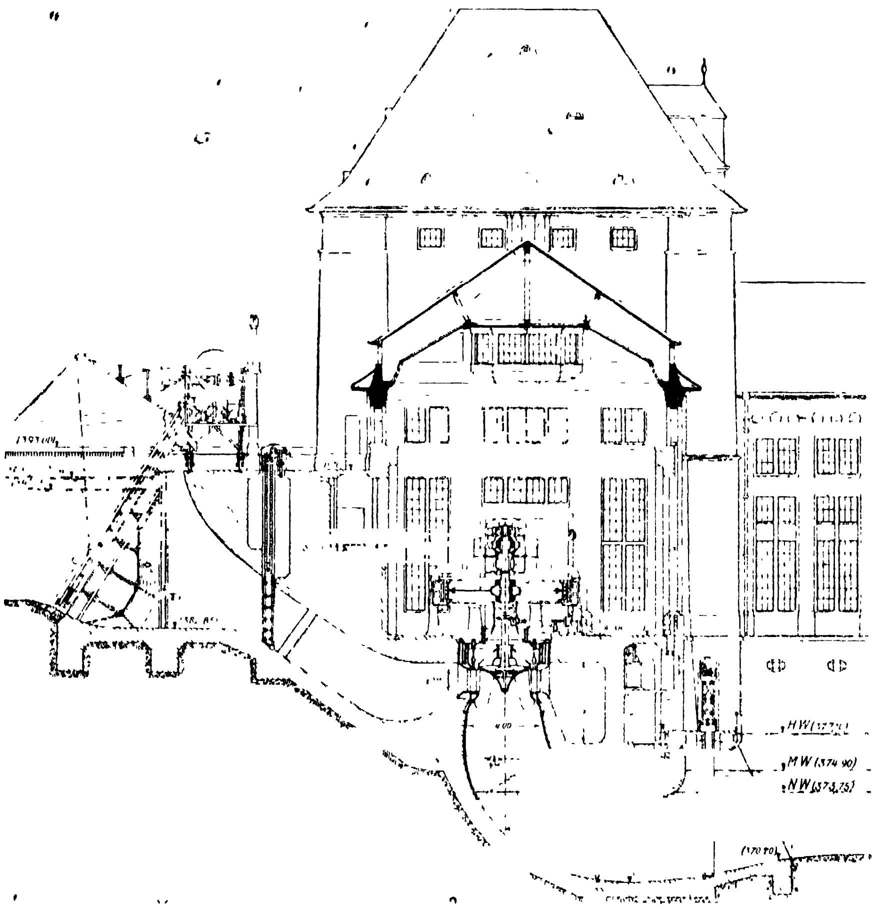


Fig. 147.—Vertical Shaft Unit at Olten Gosgen, Switzerland

For somewhat higher heads a light steel plate volute casing is provided, which is embedded in the concrete, forming the mould around which it is poured, and making it watertight. Such a design is shown in fig. 148. For still higher heads the casing may be made self-supporting as in fig. 149. In many modern installations even for medium and high heads the volute casings have been embedded in the concrete of the substructure. This gives a very stable support for the unit, and prevents all vibration, while

* By courtesy of Messrs. Escher, Wyss, & Co., Zurich.

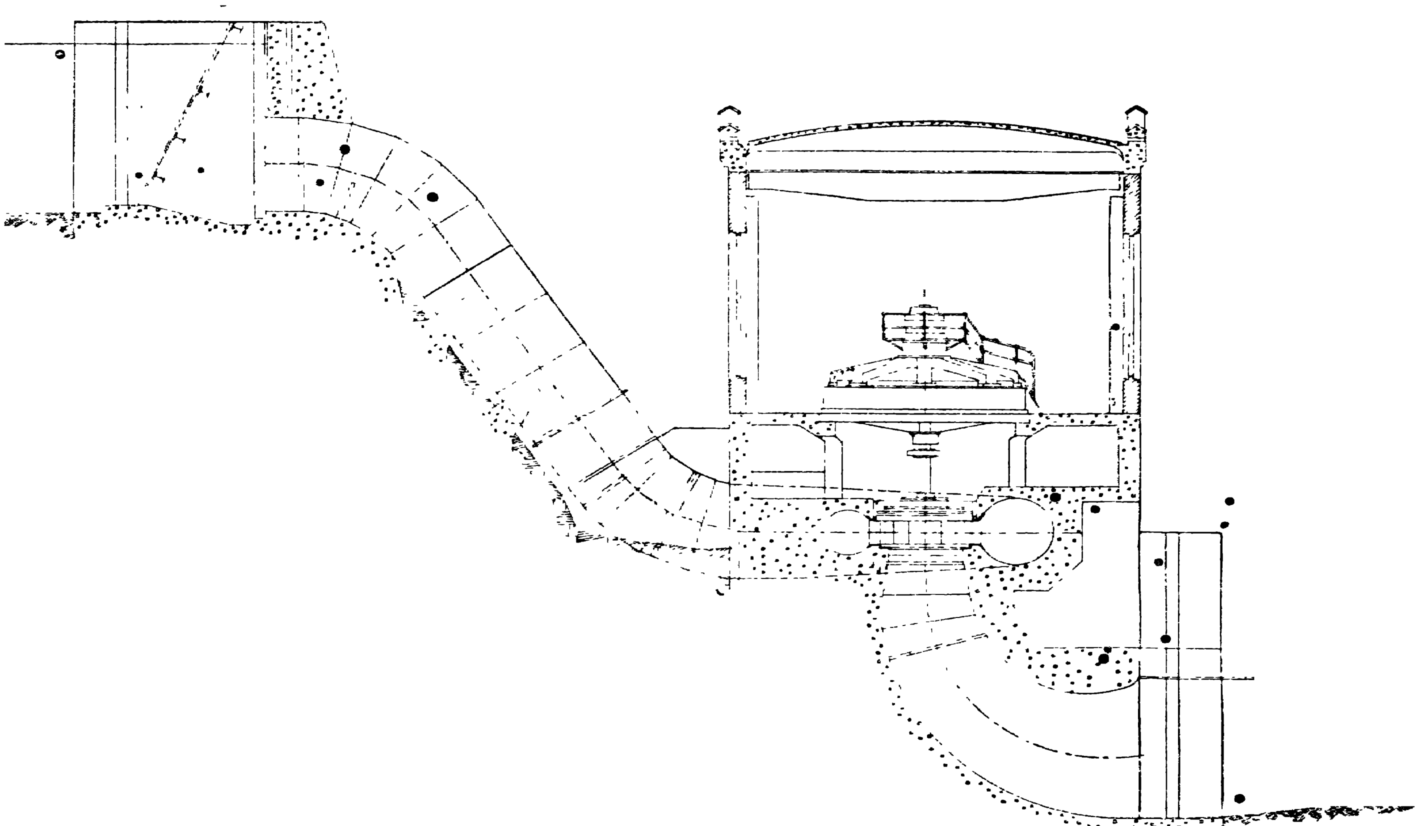
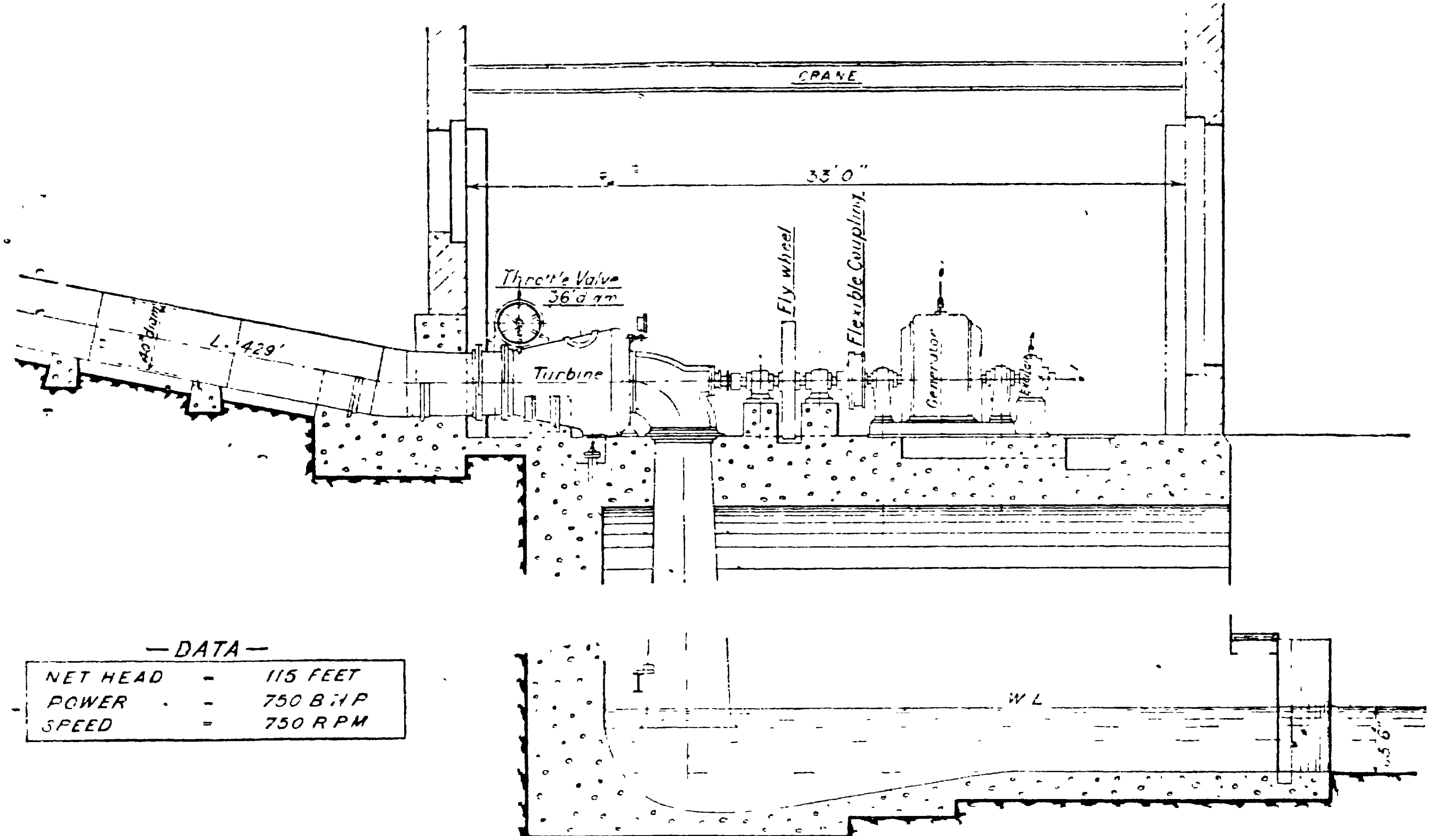


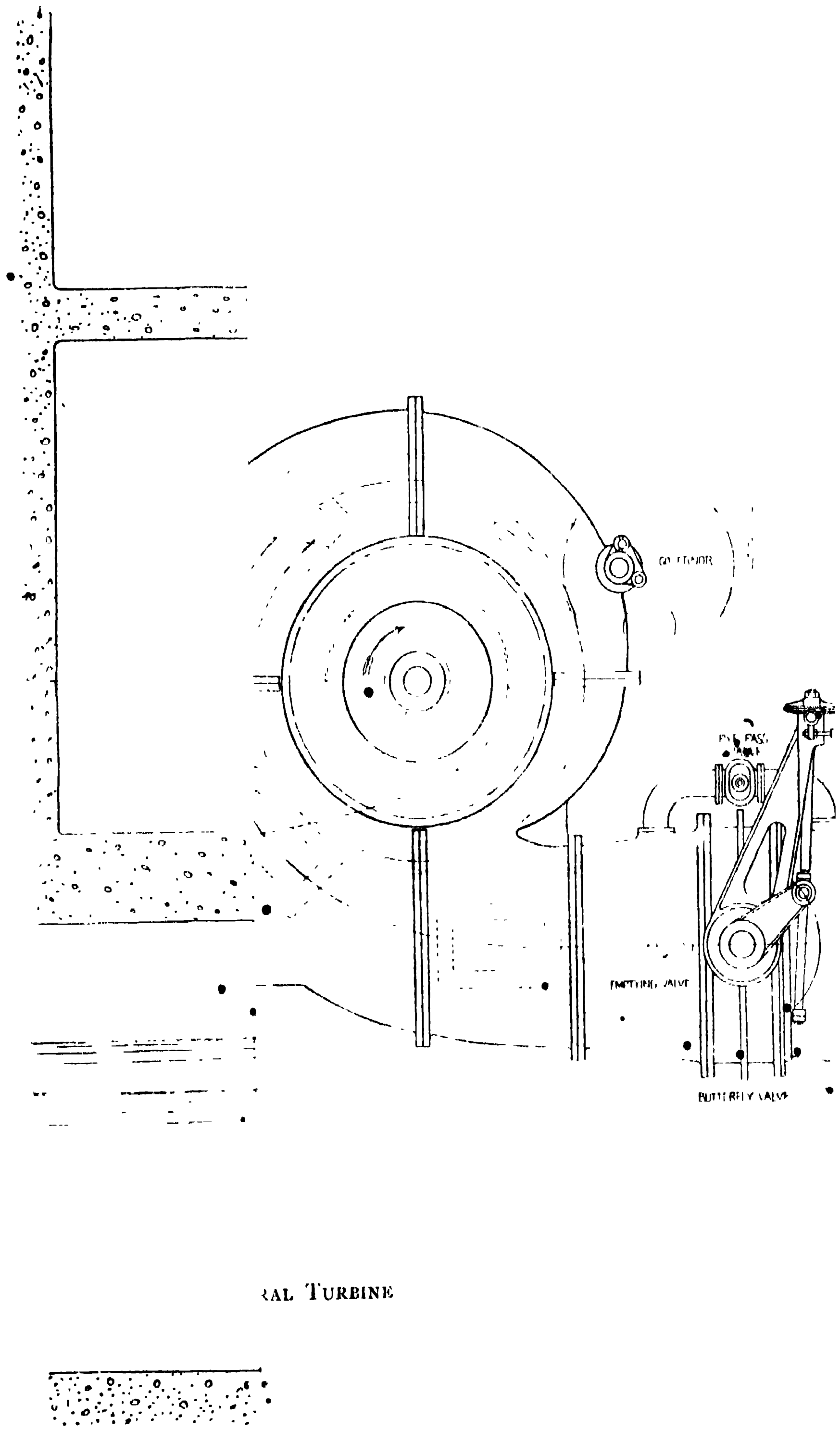
Fig. 148.—Vertical Shaft Unit with Steel Plate Volute embedded in Concrete



—DATA—

NET HEAD	-	115 FEET
POWER	-	750 B.H.P
SPEED	=	750 R.P.M

Fig 150.—Single Horizontal Turbine, with Water Inlet at end of Casing



RAL TURBINE

the mass of the concrete provides additional safety against rupture of the casing due to water-hammer shocks.

Vertical spiral units are being used to an increasing extent for heads up to 200 ft. The slight complication of the suspension bearing is outweighed by the simplification of the pipes, which come straight into the turbine without bends, and by the narrowing of the power station, which becomes cheaper and more compact.

For small units the horizontal arrangement is more common. For high heads spiral casings are general, but for moderate heads a circular casing may be used. Fig. 150 shows a single-wheel unit with its inlet at the end of the casing. This is usually of steel plate except for small units, when it may be cheaper in cast iron. The piping arrangement makes a very simple and cheap design and, as compared with the piping for a spiral-cased turbine, eliminates at

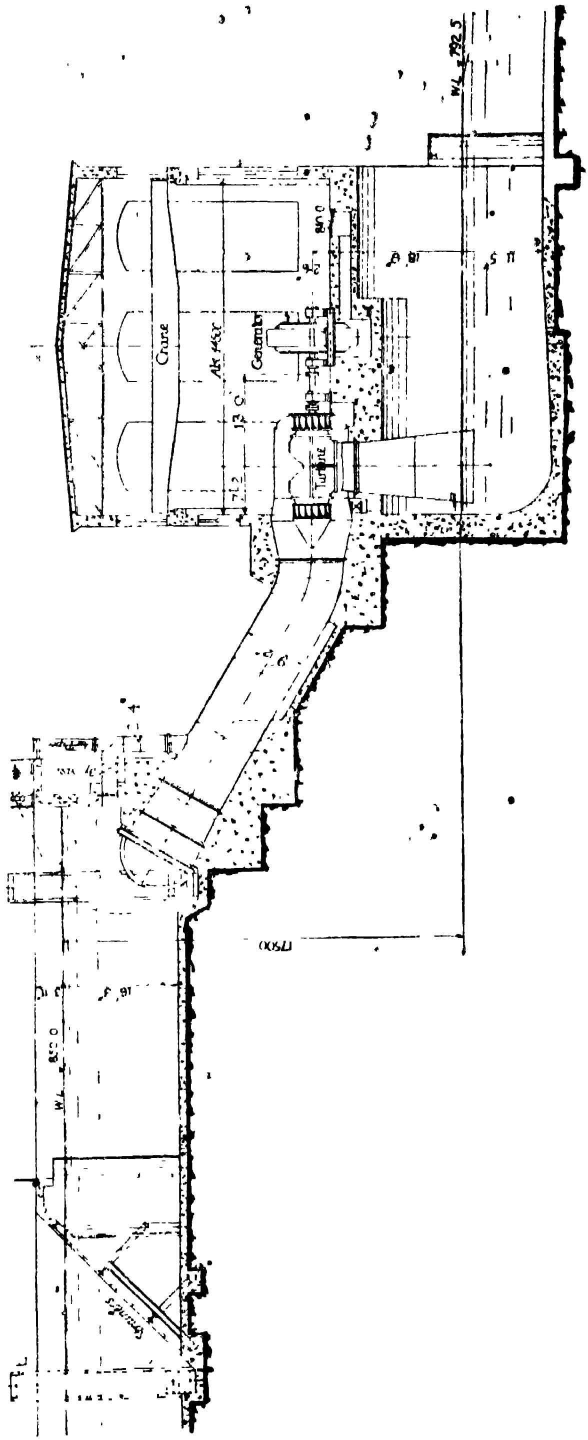


Fig 151 —Société Générale d'Entreprises Arrangement of Power House

least one bend. Enclosed turbines of this type give high efficiencies, and might be used with advantage in many cases where the dearer type with a spiral casing has been chosen. The great disadvantage of this type is the immersed bearing. If the water is not clean, the plain lignum vitae bearing cannot be used. The bearing may, however, be encased, and lubricated by a supply of clean water from a filter. Alternatively forced grease lubrication may be used, or oil-lubricated bearings designed for working under water.

In some cases the inlet at the end of the casing is not suitable, and a cylindrical casing is employed, with a radial inlet. This gives an opportunity of putting the wheel bearing outside the casing if desired.

Double-wheeled turbines are also frequently enclosed in steel plate cases. Fig. 151 shows such a unit having an axial inlet. Radial inlets are also often used. The difficulties and expedients with regard to the internal bearing are exactly the same as for the single-wheeled type of machine.

Spiral-cased reaction turbines may be used for very high heads, upwards of 750 ft. being now regarded as within their range for large units. If the water is not clean, the abrasion under such high heads is considerable, and it is better to install Pelton wheels, in which wear is not so damaging to efficiency, and in which any worn parts can easily be replaced.

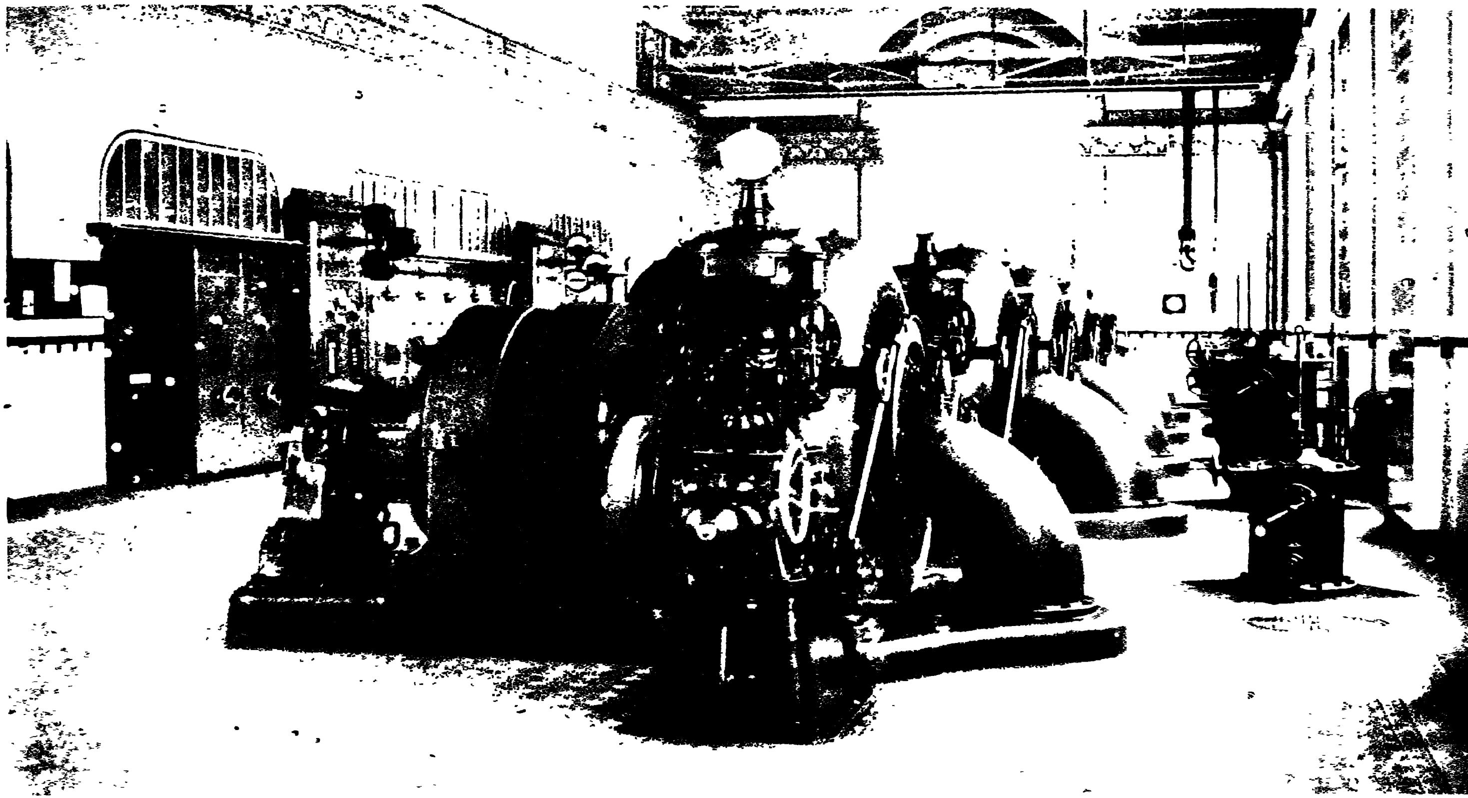
Spiral turbines are very neat, and as the pipes are usually below floor level the stations have a very attractive appearance (figs. 152 and 153).* Moreover, the bearings are all accessible, the guide-vane mechanism open to inspection, and if necessary every guide-vane spindle can be separately lubricated. They thus appeal very much to the engineer, though in many cases their extra cost as compared with enclosed turbines is not justified on this score, and as indicated above their efficiency is little, if any, higher.

Fig. 154 shows a single-wheel machine with axis across the station. A very great simplification has been achieved by hanging the turbine wheel on the generator shaft, and cutting out all but the two generator bearings. This is usually only possible when the generator itself has a considerable fly-wheel effect, so that no separate fly-wheel is necessary, though in some cases it has been achieved either by putting the fly-wheel on the opposite end of the generator shaft or between the rotor and one generator bearing. It will be noticed that a taper bend is needed to lead the water to the spiral casing from the pipe. This bend might be avoided by arranging the shaft at right angles to the pipe in the direction of the length of the station. Somewhat the same result is obtained if a single supply pipe is used, by bringing it in at the end and placing the shafts across the station, as shown in fig. 155.†. (See pocket at end of volume.)

Double spirals as in fig. 156† are used to give a higher speed for medium heads. In many cases, owing to the formation of the ground, the pipes

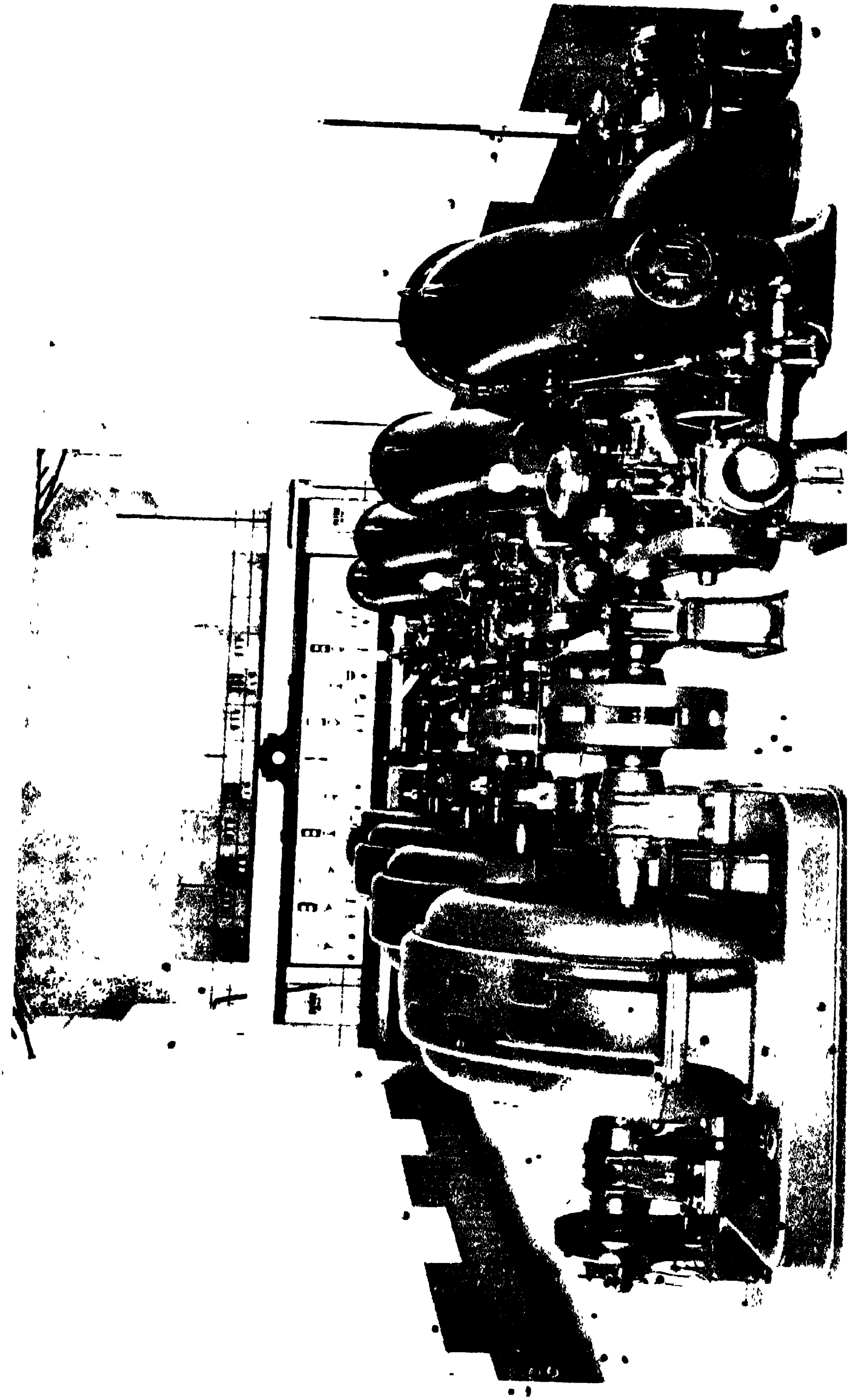
* By courtesy of Messrs. Piccard Pictet and of Messrs. Vickers, Ltd.

† By courtesy of Messrs. Boving & Co., London.



F[illegible] 152.—C[illegible] POWER STATION.

[illegible] Turbines, 350 [illegible] Head = 217 ft.



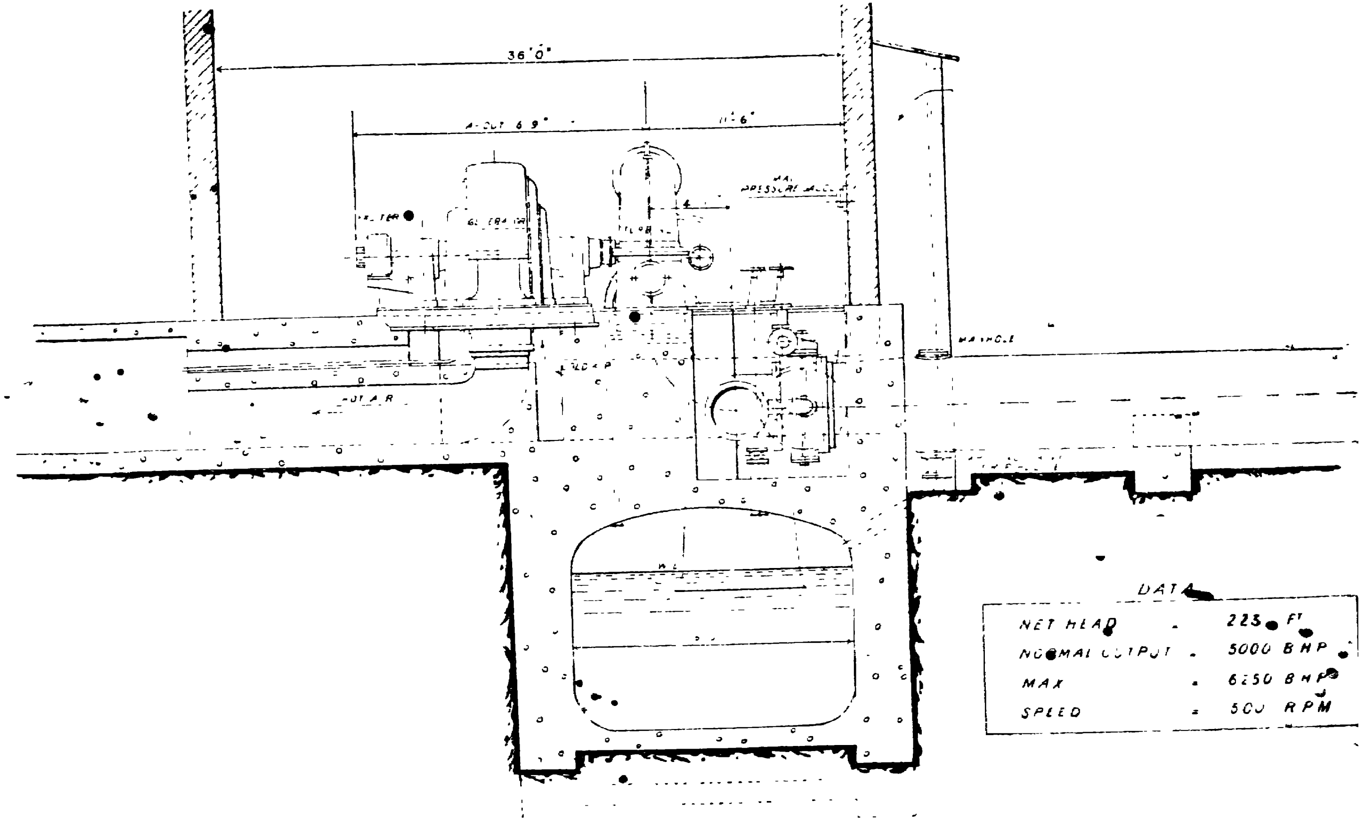


Fig 154 —Single Spiral Turbine forming a Two-bearing Unit

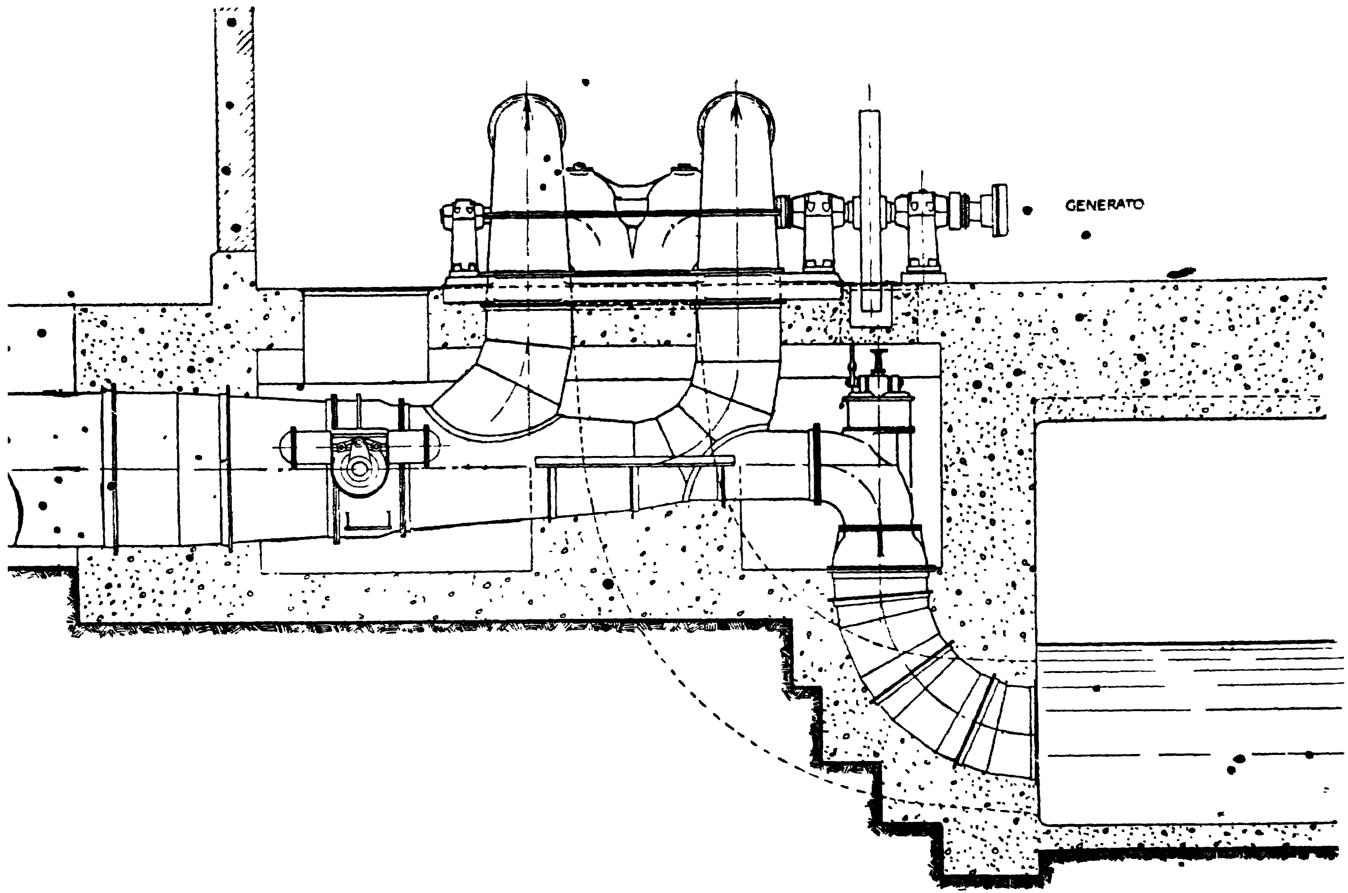
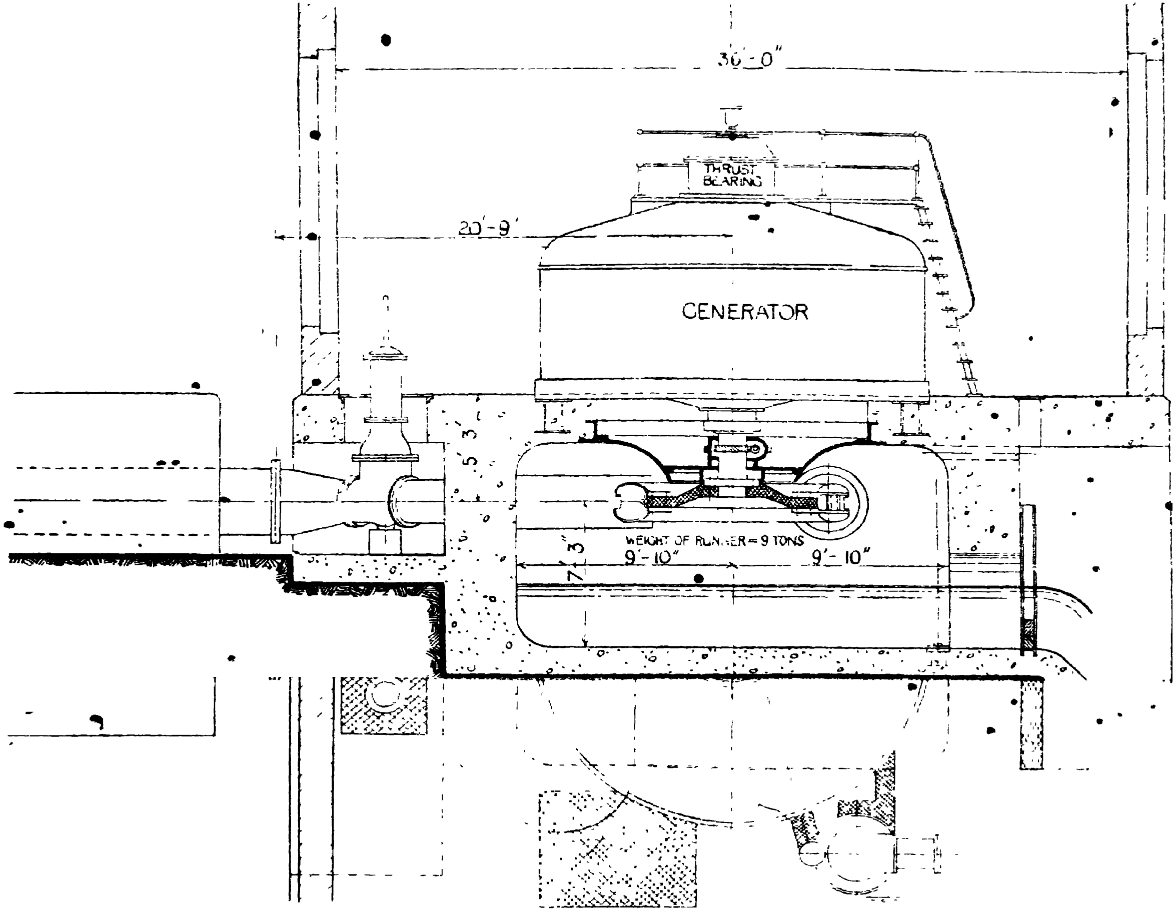


Fig 156.—Arrangement of Double Spiral Turbine with Central Discharge



30'-0"
THRUST BEARING
20'-9'
GENERATOR
5' 3"
WEIGHT OF RUNNER = 9 TONS
7' 3"
9'-10"
9'-10"

require to enter overhead, and the water is led into the cases by bends from above.

An alternative is sometimes used, in which the two wheels are placed back to back and combined in one, a single inlet and double discharge being necessary. This has the disadvantage that the bearings are a considerable distance from the wheel, and the shaft has to be stiffened to raise its critical speed sufficiently far above the runaway speed of the turbine.

The most recent practice, when double wheels are required, is to hang a turbine wheel on each end of the generator shaft, the turbine thus consisting of two separate halves, only two bearings being required for the whole combination, as shown in fig. 157. Provided the necessary fly-wheel effect can be put into the generator rotor, this forms the neatest possible arrangement.

For still higher heads Pelton wheels must be used. The selection of the number of wheels and number of jets per wheel has already been discussed in Chapter VIII.

As for Francis turbines the vertical arrangement has been favoured for large units on the score of simplicity and reduction in width of the power station. Fig. 158 gives an idea of such an arrangement. Here to get a larger power two jets have been employed. In some cases three are used, but with a greater number the loss of efficiency due to interference of the jets becomes too marked. The runner is bolted directly to the generator shaft, and no extra bearing is required. Obviously only one runner is practicable with such machines. For all but the largest units the horizontal arrangement is adopted.

The arrangement within the station is influenced by local features. Very often, in order to reduce the width of the station, the shafts are arranged in line in the length of the station. The angle of the pipe line also influences this.

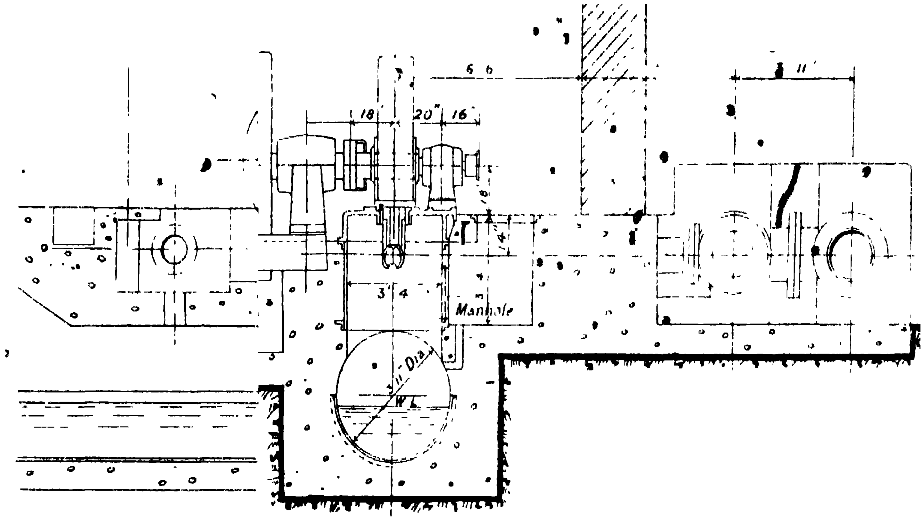
Fig. 159 shows two Pelton wheels of the single-wheel single-jet type. It will be noticed that a flange coupling is used, and that part of the weight of the runner is supported by one of the generator bearings, thus reducing the number of bearings to three. It will also be noted that the lower portion of the concrete tail race is metal lined. This is a very necessary precaution in the case of a high-head installation, in which the deflected jet may impinge with destructive effect on the bottom of this race.

In fig. 160 are illustrated two small Pelton wheels, in which the wheels are overhung so that, even though a fly-wheel has to be supported, only three bearings are required.

When four jets are needed to produce a satisfactory speed under given conditions, two runners are used each having two jets. Both runners can be placed at the same end of the shaft as in fig. 161.* Here the full number of four bearings is used, but it could be reduced to three by omitting the inside turbine bearing.

A neater arrangement is obtained by putting one runner at each end

* By courtesy of Messrs. Boving & Co., London.



…GEMENT OF PELTON WHEEL INSTALLATION. TURBINE WITH
…ATOR FORMING A THREE-BEARING UNIT.

111. Measuring Weirs.—In order to be able to obtain a definite record of the performance of the hydraulic units, and to detect any lowering of efficiency due to wear or leakage, it is very desirable to install a measuring weir as part of the initial lay-out of the plant. Fig. 163 shows such an arrangement. Here two weirs are installed in the tail race, one for each pair of turbines.

The discussion above covers the more usual arrangements of turbines of the present day, and the general considerations governing the choice have been indicated. Abnormal conditions may, however, necessitate altogether special arrangements, and each individual scheme must be considered carefully on its own merits.

112. Protection against Flooding.—Many cases of flooding, due to failure of the hydraulic equipment or excessive rise of tail water-level, have had an effect on station design. In some high-head plants equipped with horizontal turbines, the electrical bay is isolated from the hydraulic bay by a wall, and provision is made for venting the discharge, in the case of a rupture of a penstock inside the station, through the doors and windows.

It is often economically impossible to locate a power house in such a position as to be absolutely safe against all possibilities of flood due to an abnormal rise in the tail water-level at infrequent intervals, and in some cases stop log seats are provided at all doors and windows likely to be reached by such a flood.

113. As regards power-station designs, a general tendency is showing itself to confine the power-station building, where climatic conditions are favourable, to the housing of the control apparatus alone, the generating machinery, step-up transformers, and high-tension equipment being located out of doors.

CHAPTER XI

Water Power Reports

114. Compilation of Water Power Reports.—A report on any projected water power development should be as comprehensive as circumstances permit. While much depends on the immediate purpose for which the report is required, the following outline* indicates the more important features which usually require investigation and discussion.

I. *General introduction.*

(*a*) Description, including location, of site.
(*b*) Scope of investigation.

* Sensibly identical with the scheme developed by the Dominion Water Power Branch, Department of Interior, Canada. See 1914 Annual Report of the Department.

II. *Sources of data used in the report.*

(*a*) Details of personal investigation.
(*b*) Rainfall and run-off records.
(*c*) Maps.
(*d*) Existing reports

III. *Summary of report.*

IV. *Water resources of the scheme.*

(*a*) General description of the drainage area.
(*b*) Actual records, if available, showing maximum, minimum, and mean discharge for each month, also minimum for year. Measurements on site if the foregoing are not available.
(*c*) Rainfall, evaporation, and temperature.
(*d*) Storage already available, if any.
(*e*) Storage possibilities.

(1) Location of possible reservoir sites; foundations.
(2) Height of dam, and class of dam suitable.
(3) Capacity of reservoirs.

(*f*) Prior rights above or below the site; fishing rights; potable supplies; compensation.
(*g*) Ice conditions during winter months.

V. *Outline of any existing power development on the river.*

VI. *Detailed work at each site investigated.*

(*a*) Scope of inspection at site.
(*b*) Accessibility of site and transportation problems.
(*c*) Contour plan, cross-sections, and profiles.
(*d*) Foundation conditions.
(*e*) Flooding; present conditions and as modified by dam.
(*f*) Site of power house.
(*g*) Existing interests

VII. *Amount of power available.*

(*a*) With storage.
(*b*) Without storage.

In each case the maximum, mean, and minimum values are to be given.

VIII. *Estimates.*

(*a*) Cost of power developed.
(*b*) Cost of storage.

IX. *Market for power.*

(*a*) Present.
(*b*) Future, including any obvious possibilities.
(*c*) Length of any transmission lines.

X. *Suggestions and recommendations.*

XI. *Appendices.*

(*a*) Plans and photographs.
(*b*) Run-off, rainfall, and gauge records.
(*c*) Reports.
(*d*) Maps and plans of existing power plants.

Section I should cover the general features of the scheme. This involves a general description of the river and its characteristics, touching on the drainage area, the direction of flow, gradient, type of banks and of river bed, cultivation along banks, and general topographical and geological features. It should give the definite location of the site.

Section II should summarize the sources of information on which the report is founded, and indicate the route followed, the time involved, and the degree of thoroughness with which the inspection has been carried out.

In the summary of Section III, all the essential features of the report should be brought together in a brief statement, with a tabulation of the essential results where possible.

In Section IV, those features of the drainage area which are of direct importance to the question of water supply, such as the variation in seasonal flow and the probability of sudden floods, should be dealt with. The magnitude of the area should be given, along with details of the run-off when these are available. When no records are available, estimates or measurements of the flow at the time of inspection should be made by whatever method of stream measurement is most convenient. From these, in conjunction with any high-water marks in evidence and from the testimony of local inhabitants as to extreme high- and low-water conditions, as careful an estimate as is possible should be made of these conditions. The times of year at which extreme high and low water are usually found should be given.

The rainfall as recorded at the nearest or most suitably situated gauging stations should be discussed, and should be utilized in an estimate of the run-off if no stream flow records are available. Evaporation records, if available, should be considered in this connection. Temperature records are useful as enabling some approximation to the probable evaporation to be made failing any direct measurements.

The question of storage possibilities should be covered as thoroughly as circumstances permit. Any suitable dam site should be investigated, and a note made as to the foundation conditions, the most suitable type

of dam, and of its height and length. The extent of ground covered, with various amounts of storage, should be ascertained. The capacity of the storage reservoir, together with the area of the adjacent catchment areas and their sufficiency to fill the reservoir, should be fully covered, and the beneficial effect of such storage on the flow of the river should be discussed. Mention should be made of any existing or projected schemes of municipal water supply, irrigation, or water power which might affect the amount of water available.

In a non-temperate climate the general ice conditions in winter should be determined and discussed. The conditions to be anticipated at the site as regards frazil and anchor ice, the possibility of ice jams both above and below the site, and the effect of these on the head- and tail-water levels should be noted.

In Section V, existing power developments on the river likely to be in any way affected by the proposed scheme, should be briefly discussed under the heads: ownership of the plant; when constructed; brief description; output and load factor; probable effect of suggested development on its operation.

Section VI. When no definite scheme of development has been proposed, the inspecting engineer is expected to outline the scheme which his study at the site may suggest as being most feasible. He should gather all the information and field data which may be essential to a proper consideration of the scheme and of its cost. A provisional lay-out should be shown on the contour plan of the site. Arrangements shall be made on the ground for the installation and reading of stream gauges at all points where such records are desirable.

The accessibility of the site, including the distance to the nearest rail and water transport, the ease or difficulty of constructing a branch line, the condition and suitability for heavy transport of the roads in the vicinity, and in short the best means of connecting the site with existing lines of traffic, should be discussed.

Enough rough instrument work must be done to permit of plotting a fairly accurate contour map of the vicinity covered by the proposed lay-out. The plan should indicate any outcrops of rock, and any clay, sand, gravel, loam, &c., which may be in evidence Water-levels at important points, with the date of observation and a note as to the state of the river, should be plotted on the plan. The high- and low-water levels to be expected in the tail water of the projected scheme are of special importance.

A cross-section of the river bed and both banks along the site of the proposed dam should be plotted. This should indicate the character of the ground surface and of the river bed, and of foundation conditions either in evidence or assumed. The records of any borings are to be given. A profile of the river surface, and if possible of the bed from a point upstream from the dam below the tail race is required. A profile section through the dam, intake, head race, canal (or pipe line); power plant, and tail race, showing such essential elevations as head water, crest of dam, tail water,

&c., should also be given The plans should show the extent of the ground at present liable to inundation in time of flood, and the extent of the area which would be covered by the construction of the storage reservoirs. The site of the power house should be indicated, with a note as to any difficulties likely to be experienced in its construction or in that of the penstock, forebay, or tail race.

Any existing interests such as roads, bridges, or buildings which may be affected by the construction of the plant and the consequent flooding should be indicated. The question of fishing interests, logging or navigation rights, and of possible compensation water should be discussed.

In Great Britain the largest scale maps, published by the Ordnance Survey of many of the districts of interest in connection with water power developments are to a scale of 6 inches to the mile, which is sufficient to satisfy standing orders for Private Bills, when application is made to Parliament for powers of compulsory purchase. This scale is very suitable for showing the various works and the important contours. When the flow of any river is intercepted, standing orders require that the drainage areas from which it is to be diverted should be shown to a scale of 1 inch to the mile. In the case of any works it is necessary to deposit sections to a scale of not less than 4 inches to the mile horizontal and 1 inch to 100 feet vertical.

In Section VII, the possibilities of power developments both with and without storage should be considered. In the latter case the power available with the minimum flow, and the power which might be developed during the eight or nine months not including the extreme low-water season should be discussed. Consideration should be given to the possibility of developing the power more cheaply from a steam or gas installation. The question of utilizing existing steam or gas plants as a reserve, or as auxiliary to the proposed water-power plant, or of installing such a plant as part of the proposed scheme should also be covered.

In Section VIII, approximate estimates of the cost of the proposed scheme, and the basis on which these are made, should be given. These estimates should show separately the costs of the land and water rights, the civil engineering construction, and the hydraulic and electrical machinery. Approximate figures of the annual charges, including interest, sinking fund, depreciation, rates and taxes, repairs, supplies, wages and salaries, should be given, and the cost per unit of output calculated, both on the assumption of a continuous output and of an output corresponding to an ordinary industrial load. This should be done for the alternative schemes if more than one is put forward.

Section IX will involve as thorough an investigation as the circumstances warrant of the present and future power market in the surrounding district. Possibilities of the development of special local industries based on the natural resources of the district, whether mineralogical or otherwise, should be noted, along with the possibilities of economical power transmission to more remote industrial localities. In the latter case an estimate

should be given of the cost of such transmission and of the power available at the end of the line, as well as of any possibilities of disposing of power *en route*. Any suggestions or recommendations with reference to the foregoing points should be set out in full. The location of suitable gauging stations for the continuous record of river flow should be covered.

Where more than one site is investigated, the relative advantages should be discussed, and a definite recommendation made as to their relative advisability.

In general, a preliminary report should make it quite clear whether the scheme is one which, from an engineering and a financial standpoint, is reasonably feasible, while a final report should give sufficiently detailed data to enable any financial body definitely to determine the ultimate prospects of the scheme.

APPENDIX

The Hydraucone and Spreading Draft Tubes

In the conventional theory of turbine design it has been usual so to design the vane angles at exit as to discharge the water without any appreciable tangential velocity. Even with a slow-speed high-head turbine however, this can only be the case with one gate opening, and with a modern high-speed runner there is a considerable velocity of whirl at all gate openings.

It is usual to assume that this whirl is analogous to that in a free vortex, i.e. that its velocity is inversely proportional to the radius, and that the pressure increases radially outwards according to the law

$$\frac{p_1 - p_2}{\omega} = \frac{w_2^2 - w_1^2}{2g} = \frac{\omega^2}{2g}\left(\frac{1}{r_2^2} - \frac{1}{r_1^2}\right),$$

where w represents the velocity of whirl; ω the angular velocity; and r_2 and r_1 are two radii of which r_1 is the greater.

Thus if r_1 and r_2 are the radii of a straight conical draft tube at the points of exit and of entry, the gain of pressure due to the conversion of the kinetic energy of whirl will be given by the above expression, or more correctly by

$$k\frac{\omega^2}{2g}\left(\frac{1}{r_2^2} - \frac{1}{r_1^2}\right).$$

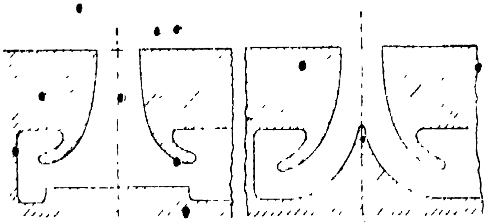
There is little definite data as to the value of the coefficient k, but it is probable that in a draft tube of moderate dimensions it is approximately 0·75.

In a low head plant it is often necessary to install the turbine so near to the tail race level that a straight conical draft tube giving a reasonably low velocity of discharge would be very inefficient owing to the large angle of flare required, and in many such cases bent draft tubes have been used. While such a tube enables any suitable length and angle of flare to be adopted without excessive excavation, it suffers from the disadvantage that the kinetic energy of whirl is dissipated in eddy formation, and is therefore not to be recommended where such loss is likely to be serious.

With a view of overcoming this difficulty, two types of straight draft tube have been developed during recent years, in which the water at the point of discharge from the tube is constrained to flow radially outwards between two surfaces approximately parallel and of considerable radius;

thus giving a very low velocity to both the radial and tangential components of the discharge, and at the same time enabling the water to be discharged horizontally in the direction of flow in the tail race.

These types are known respectively as the " hydraucone regainer " and the Moody " spreading draft tube ". In the former the water is discharged against a flat plate which deviates its direction (fig. 165(*a*)), while in the latter



(*a*) (*b*)

Fig. 165

a conical surface is provided for the same purpose (fig. 165(*b*)). Since in a free vortex the pressure is a minimum at the centre, cavitation effects will always be experienced in the first place at the centre of the draft tube immediately beneath the runner, and to prevent this in some recent low head plants this conical surface has been continued up the draft tube as far as the runner.

Such designs have the advantage of enabling a large ratio of outlet to inlet area to be obtained with a comparatively short tube.

Useful Data

1 cubic foot of water weighs 62·4 lb.
1 gallon (imperial) weighs 10 lb.
1 gallon (U.S.) weighs 8·3425 lb.
1 imperial gallon is equivalent to 1·2 U.S. gallons.
1 second-foot, or 1 c. ft. per second = 6·24 imperial gallons per second.
1 second-foot = 31,536,000 c. ft. per year.
= 1·983 acre-feet per day.
= 55·54 acre-feet per month of 28 days.
= 57·50 ″ ″ 29 ″
= 59·50 ″ ″ 30 ″
= 61·49 ″ ″ 31 ″
1 acre-foot = 271,814 imperial gallons.
= 43,560 c. ft.
1 inch deep on 1 square mile = 2,323,200 c. ft.
1 acre = 43,560 sq. ft. = ·4047 hectare.
1 sq. mile = 2·59 sq. kilometres.
1 c. ft. = ·0283 c. m.
1 horse-power = 550 ft. lb. per second.
= 746 watts. = 0·746 kilowatt.
= 76·0 kilogram-metres per second.
1 kilowatt = 1·34 horse-power.

With an efficiency of 80 per cent, cond-foot develops 1 h.p. under a head of 11 ft.

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